MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

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Los Angeles, Cal., April 16-18

Spring Meeting

Montreal, P. Q., Canada, May 28-31

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Contributors and Contributions

Large Machine Tools; Design and Construction

George H. Benzon, Jr., outlines in this issue the general character of the work of design and construction of large machine tools. This paper was one of a group of papers on machine tools presented at a recent joint meeting of the Philadelphia Local Section and Machine-Shop Division of the A.S.M.E. the Engineers' Club of Philadelphia and the Philadelphia Section of the A.I.E.E.

Mr. Benzon has been associated with Wm. Sellers & Co., Inc., Philadelphia, since 1898 when he served his drafting-room apprenticeship with that concern. In 1904 he was put in charge of planer design and construction, and four years later became chief draftsman. Since 1919 he has held the position of engineer.

Efficiency of Scotch Marine Boiler

In his paper dealing with this subject, C. J. Jefferson shows what has been effected by careful operation and provides a definite target of boiler efficiency at which marine engineers may aim with profit. Mr. Jefferson is a 1910 graduate of Cornell University. For seven years he was associated with the American Ice Co. as testing and mechanical engineer. During the War he served as a lieutenant in the U. S. Navy. Upon his discharge he took charge of the Boiler Unit, Technical Section of the U. S. Shipping Board. At present Mr. Jefferson is head of the Fuel Conservation Section of the U. S. Shipping Board.

Effect of Pulsations on Flow of Gases

This paper, by Prof. Horace Judd and D. B. Pheley, discusses work undertaken under the joint direction of the engineering Experiment Station of Ohio State University and the A.S.M.E. Research Sub-Committee on Fluid Meters. Professor Judd was graduated from Ohio State University in 1897 with the degree of M.E. and then served for two years as an instructor in the mechanical engineering laboratory, receiving his M.S. at the end of that period. For three years he held a similar position in Pratt Institute, Brooklyn, and then returned to Ohio State University to serve as assistant professor of experimental engineering until 1910 and as associate professor until 1920, when he was appointed professor of hydraulic engineering.

Mr. Pheley is at present junior engineer in the U. S. Coast and Geodetic Survey. He was graduated from Ohio State University in 1921 as a civil engineer and during that summer served as assistant to Professor Judd in the study of and experimental work in the problem of pulsating flow.

Engineering Aspects of the Design of Musical Instruments

The specific problems presented in the manufacture of pianofortes are considered by William Braid White, who shows that vast improvements would be achieved if the principles of mechanical engineering were more generally adopted as the foundation of such manufacture. Mr. White, an Englishman by birth, was educated at St. Paul's School and King's College, London. He came to the United States in 1896 and became interested in piano making. Since 1904 he has been technical editor of the *Music Trade Review*, New York; he is also associate editor of the *Talking Machine World*, New York.

Airship for Long-Haul Heavy-Traffic Service

The factors upon which the value of an airship as a carrier depends form the subject of this paper. Ralph H. Upson, the author, is chief engineer of the Aircraft Development Corporation of Detroit. He was graduated from Stevens Institute of Technology in 1910 with the degree of M.E. and entered the aeronautic department of the Goodyear Tire and Rubber Co., Akron, Ohio. He designed and built his first complete balloon in 1912, and in 1913 won the national and international balloon races. The Wingfoot Lake Flying School was started by Mr. Upson in 1917. He won national balloon races in 1919 and 1921, served on the Navy Department Commission in Europe, 1918–1919, and studied European methods in 1920. From 1915 to 1920 he was chief engineer of the Goodyear aeronautical department.

Hydraulic-Transmission Variable-Speed Drive

The progress that has been made in applying "hydraulic transmission" variable-speed drive to machine tools and ordinary manufacturing processes is reported in this issue by Walter Ferris. Mr. Ferris was graduated from Lehigh University as a mechanical engineer, class of '95. Until 1900 he was employed by the Pencoyd Iron Works in the master mechanic's office, and by the Laflin & Rand Powder Co. in charge of the design and erection of powder mills. From 1902 to 1921 he served the Bucyrus Co., South Milwaukee, Wis., as draftsman, chief engineer and consulting engineer, consecutively. The Oilgear Co., of which Mr. Ferris is vice-president, was organized in 1920.

Safety Engineering in Compression of Gases

This paper outlines a few of the chief hazards that are associated with the compression of some of the gases in common use in industry. Its author, A. D. Risteen, was graduated from Worcester Polytechnic Institute in 1885 with the degree of B.S. He received his Ph.D. from Yale in 1903. For twenty-three years he was associated with the Hartford Steam Boiler Inspection & Insurance Co. For the last ten years he has been in the engineering division of the Travelers Insurance Co., and is now director of technical research and safety publication work for that concern and the Travelers Indemnity Co.

A.S.M.E. Pacific Coast Regional Meeting

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A.S.M.E. Spring Meeting, Montreal, May 28-31, 1923

MECHANICAL ENGINEERING

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Design and Construction of Large Machine Tools

Limitations Imposed on Design by Materials and Available Shop Equipment—Problems Involved in the Design and Construction of Large Boring Mills and Planers

By GEO. H. BENZON, JR.,[1] PHILADELPHIA, PA.

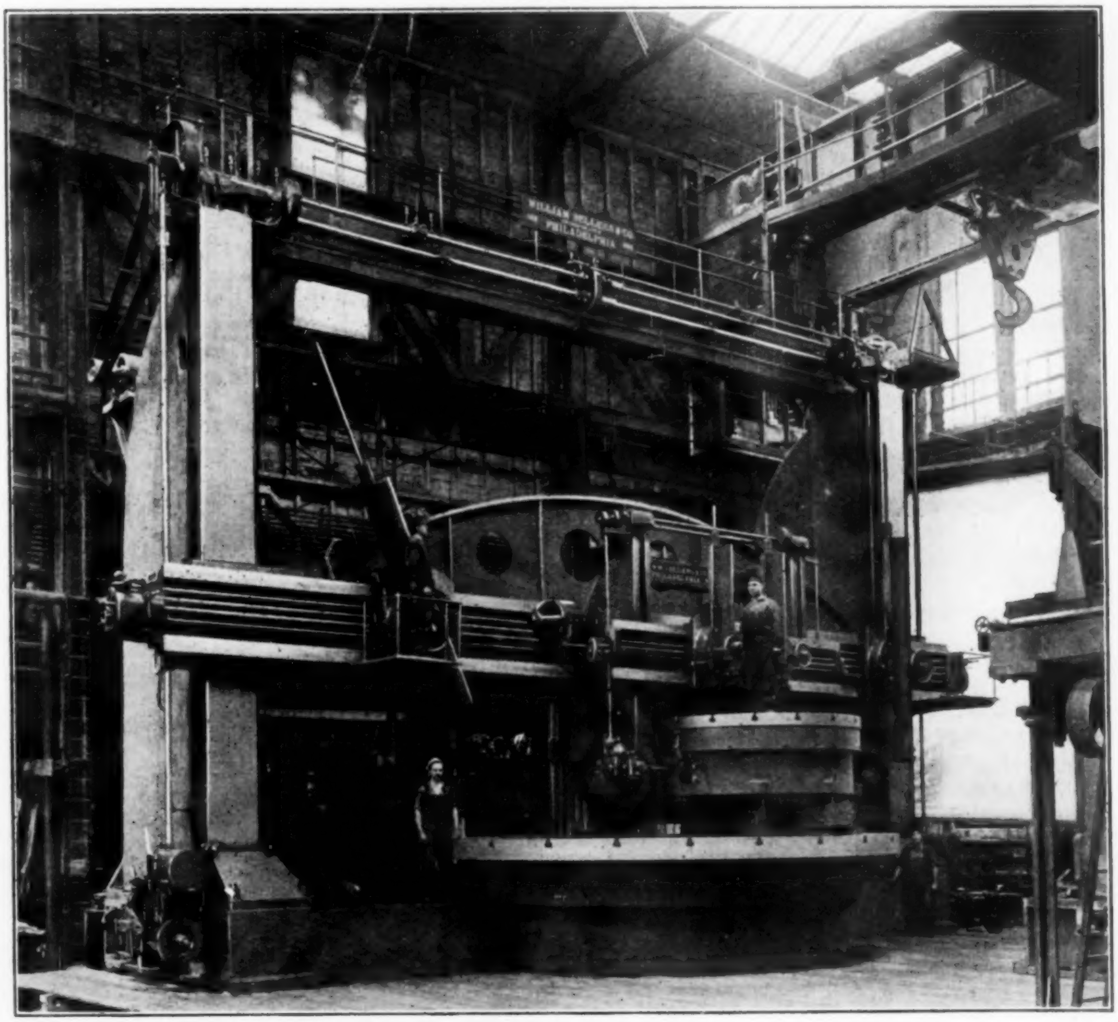

FIG. 1 LARGE BORING AND TURNING MILL

(Swing between uprights, 35 ft.; clearance between work table and underside of tool head, 18 ft.; total weight, 429,200 lb. Note small mill set on work table of large mill; swing between uprights, 100 in.)

IT HAS BEEN difficult in preparing this paper to sort out from the mass of every-day facts those points which might be of interest in a general discussion such as seems to be indicated by the title selected. The subject as generally understood is a very broad one. In fact, it covers two somewhat different types of machines commonly classed under the general title. There are machines which are used for accurate cutting and finishing; these belong distinctly in the machine-tool class. There are also those which are used for metal working other than finishing. While these may be, and are, often considered to be in this class, there is no sharp line between them and the machines used for metal working which are grouped under other machine divisions. The first-mentioned class, however, will be the one made the subject of this discussion.

The shops building hydroelectric machinery, marine engines, blowing engines, and rolling-mill machinery have been the main sources of demand for large machine tools. Ordnance and battleship development during the late war created demands that bade fair to swamp the large machine-tool builders, but these subsided and dropped out entirely after the cessation of hostilities. Large marine Diesel engines, if commercially developed in this country, will probably widen further the market for large machine tools.

It is obvious and natural that the demand for large tools di-

[1] Engineer, Wm. Sellers & Co., Inc. Mem. Am.Soc.M.E.

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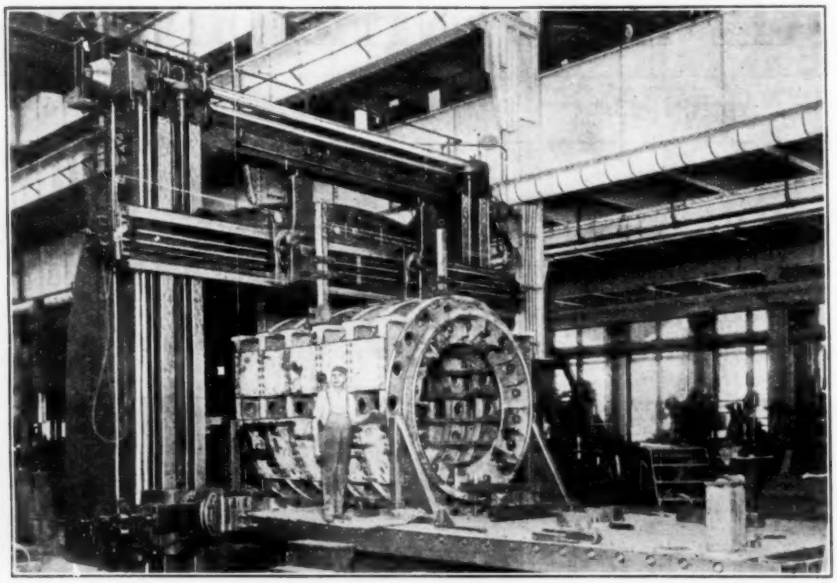

Fig. 2 Large Planing Machine

(Length of table, 38 ft. 6 in.; clearance between housings, 16 ft. 2 in.; distance from table to underside of crossrail, 13 ft. 4 in.; overall length of bed, 65 ft. 8 in.; total weight, 412,340 lb.)

minishes as the sizes of the tools increase. Developments of large mechanical and electrical projects of continually increasing size must include consideration of the means of production, and involve in many cases installations of large machine tools to accomplish the ends desired. The development of one such project frequently raises the general requirements in the particular industry to which it belongs and incites competing development. This may account in part for the spasmodic markets for the different lines of large machine tools. The requirement for these tools is very irregular. Some lines lie dormant for a long time, during which others are very active, and then the active ones drop out entirely for a period and others take their place. There are also times when the saturation point is temporarily reached, during which the whole market is dead.

From the foregoing it can readily be understood that large machine tools cannot be constructed or handled as a continuous manufacturing process. Consideration of this fact affects all branches of the industry from the design to the final inspection. The market has been so irregular as to make it seem inadvisable or uneconomical to equip and maintain establishments for the sole purpose of building such tools.

Machinery for building in the most efficient way the ever-increasing sizes of tools the market demands, would involve an investment far in excess of that which would predicate an equitable return. The machine-tool builder therefore in many instances has to use average large-shop equipment to produce tools of a larger size than any he has in his establishment. This condition requires considerable attention from the designer, who must plan the large units of the machine so that they may be handled by existing equipment and at the same time not suffer in their value as a part of the big machine tool.

Limitations Imposed on the Designer

Large patterns are bulky in storage and expensive in maintenance—to such an extent that in many cases the designer must try to adapt them to a variety of uses. It frequently pays to make a mold from a larger pattern and to stop it off with cores, rather than make a new pattern for the job. The time required to make new large patterns for a machine might affect delivery dates seriously and cause the loss of sales. The designer must bear all of this in mind, and also consider the feasibility and desirability of compromise in shaping the large units.

Another factor entering into the design and construction is the limiting sizes of parts that can be conveniently shipped. Bridge and tunnel clearances on the railroads limit the dimensions of parts that can be shipped as a unit. Machine tools must be made so that after having been assembled, tested, and corrected where necessary to an accuracy seldom matched in other mechanisms, they may be taken down, shipped, reassembled, and reproduced after these operations the same degree of accuracy as when under test.

The quality of machine tools for finishing is dependent on sound design backed up by expert foundry and machine work; and all of these factors must rest on the foundation of experience—not only that of the individual but that which lies in the records of an establishment. The grade of mechanic required in building these big tools is much higher than the average, the valuable men being those who have had a number of years of experience in this class of work. Detailed planning and instruction can rarely be applied in such work, and therefore the individual experience of the men, as well as of the foremen, has an appreciable effect on the character of work produced.

In the drafting room, as has been noted, there must be a thorough knowledge of shop conditions, shipping conditions, and of the difficulties encountered in previous experience in building the big machines. In the pattern shop and foundry there must be knowledge of the action of the molten metal in cooling, which will enable the patternmaker and the foundryman to gage with some degree of certainty the eccentricities of the metal in large castings so that those delivered to the machine shop will be sound, the bearing surfaces will show a good close-grained material, free from sand or casting defects, and the finished surfaces will have such hardness as will produce good wearing qualities and at the same time will not be too hard to machine satisfactorily without undue trouble with the cutting tools.

A large casting must be designed so that as far as possible the

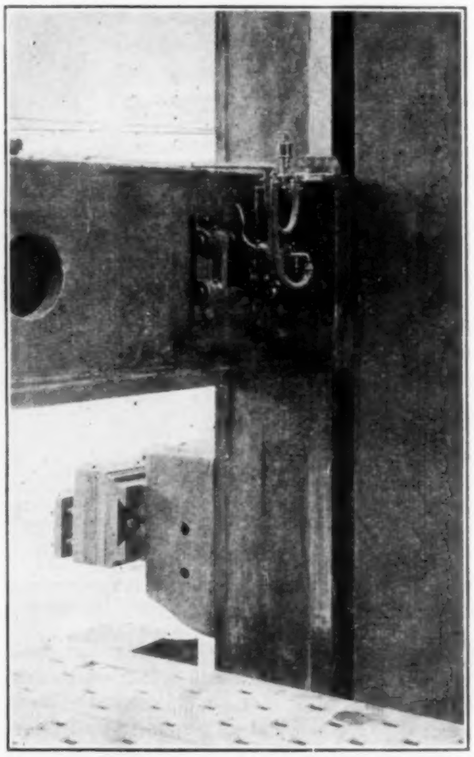

Fig. 3 Pneumatic Device for Clamping Planer Crossrail

various sections will balance each other in cooling. Lack of balance in a long casting, for instance, if not properly compensated, will cause it to distort in cooling, often to such an extent that the amount of finish allowed in the pattern shop is of no use. There are some cases, however, where it is necessary to use unbalanced sections. In these cases experience teaches how much the mold should be distorted to offset the natural tendency of the casting. Big castings for machine tools require the handling of large quantities of molten

metal in the very short time that can be allowed for pouring even the largest of castings. Interrupted flow of metal in pouring a large casting may, in fact will, if it occurs, detract from the value of the casting.

In order to produce a good close-grained material on the wearing surfaces of the machine, it has been found desirable and practical, using a high-silicon iron, to cool these surfaces quickly by casting them against chill blocks. The metal produced in this way is not what is known commercially as chilled iron. It is easily machined and not extremely brittle. While it is common knowledge that this method is desirable and practical, some years of experience as to size of chills and analysis of metal are essential in the making of large castings in order that satisfactory and consistent results may be obtained.

In the machine shop the castings must be properly handled to produce accuracy. All castings as they come from the foundry contain a great number of internal stresses. When some of these stresses are released in machining, others that they have been balancing are also released and produce distortion. It is therefore necessary that these large castings should be rough-machined in the proper sequence of operations and that enough time between be given for the stresses to work themselves out before final conditioning.

One of the disturbing factors in the construction of large machine tools is the variation in temperature between night and day and between floor temperatures and temperatures which exist at a height of, say, 25 ft. above the floor. For instance, in testing the alignment of the housings or uprights on large machine tools, variation in temperature will cause uprights to move from one side of a plumb line to the other in comparatively short time. Due allowance for variation in temperature, timing of the tests and inspection, during the course of erection, to fall under proper conditions, are very necessary in building these machines, particularly as such an upright must be plumb to within very close to a thousandth of an inch in its whole length.

The main bearings on large machine tools must be accurately fitted. To do this conveniently, where possible the bearings are capped and bushed and each bearing scraped to fit its individual shaft. The shaft bearings are ground. In fitting the shafts to the bearings on large machine tools, shoulders are turned on the shafts

Fig. 4 25-Ft. Boring and Turning Mill with 10-Ft. Stroke of Tool

(The crossrail here is so deep vertically to meet requirements that the reinforcing arch girder is omitted satisfactorily. Total weight of machine, 373,130 lb.)

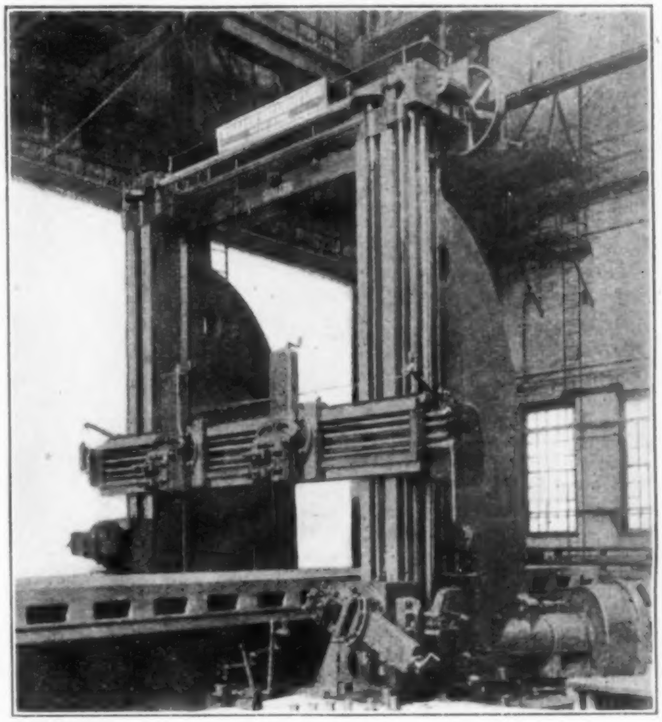

Fig. 5 12-Ft. Planing Machine

(As in the mill of Fig. 4, the crossrail is so deep vertically that a reinforcing arch girder is not needed.)

to suit. Measurements from the stands and housings after they are set up and bolted to place, are transferred to the shafts with proper allowances for running clearance. A properly designed bearing, if improperly fitted during erection, may in a very short time of running develop such trouble that the design of the bearing may be questioned and throw discredit on the machine performance.

All of the main bearings in high-grade machine tools are bushed in order that (1) the best possible kind of metal may be provided for the bearing surface, and so that (2) in case of accidental cutting or scarring or wearing in service the bearing may be renewed by replacing an old bushing with a new one. In order that replacement may be properly effected the bushings are first bored and then turned on a mandrel so that concentricity of their inner and outer surfaces may result. This insures that a replaced bushing will restore correct alignment. The practice of boring bushings after they have been fastened in the stand is bad, and not tolerated by reputable builders.

Illustrations of Problems Confronting the Designer of Large Machine Tools

A detailed discussion of design and construction as applied to any one tool, to be complete, would take up more space than is available for all of this paper. However, one or two specific cases may be taken to illustrate some of the outstanding points.

Fig. 1 will give an idea of the proportions of a large boring and turning mill. To appreciate the size, comparison may be made with the medium-sized boring mill set on the work table of the large mill. The small mill shown swings 100 in. between the uprights or housings; the large mill swings 35 ft. and has a maximum clearance between the work table and the underside of tool heads of 18 ft. The rams at whose lower ends the cutting tools are carried have a stroke of 7 ft. The weights of some of the large castings are as follows: bed, 98,000 lb.; table, 73,000 lb.; crosshead bar, 66,000 lb.; total weight of machine, 429,200 lb.

The main problems confronting the designer in this instance were that the machine was to carry on the table work weighing up to 75 tons, bore a true hole 7 ft. long, and face and turn accurately. The accuracy required in the bore will probably not permit over 0.002 in. variation in round or in parallel in the length of 7 ft. To obtain the requisite accuracy the guides on the crossrail on which the tool heads move must be very close to a perfect parallel to the annular bearing surface on top of the bed. Obviously the tendency

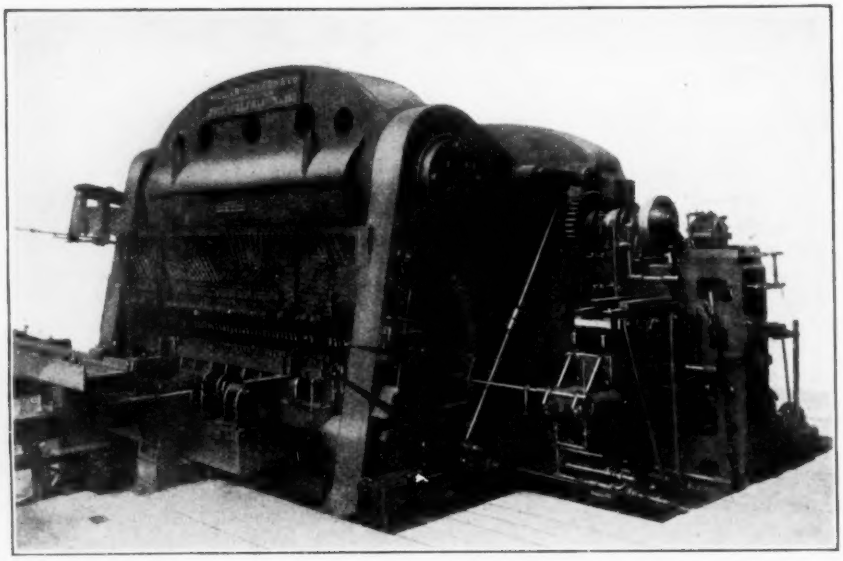

Fig. 6 Multiple Punching Machine. Operated Automatically from a Punched Paper Templet

(Used for duplicating layouts and very accurate in spacing. One of these machines has punched 59,600 holes in 9 hours. It may be broadly classed as a large machine tool.)

of a beam suspended on screws 36 ft. from center to center is to deflect somewhat, due to its own weight and the weight of the tool heads. To remove the sag in the crossrail, a separate casting in the form of an arched girder is bolted to the top of the crosshead bar. The arched girder is mounted between abutments at the two ends of the crossrail, in between the uprights and in a vertical plane as near to the front edge of the crossrail as saddle clearances permit.

Rigid Support for Cutting Tool

Another problem confronting the designer is that of supporting the cutting tool in a sufficiently rigid manner to take reasonably heavy boring cuts when the tool is extended, say, 7 ft. to 10 ft. below the bearing. This puts a bending as well as a twisting strain upon the tool bar, and a twisting and bending strain upon the crossrail. To resist these forces in the crossrail, the rail is made of a box construction. In the machine shown in Fig. 1 this box is a continuous section the full distance between the two uprights and is clamped at the back edge of this extended section to the uprights, at both sides. It is much easier to design a sufficiently rigid crossrail along these lines than in the case where the curved-back crossrail is used, in which the box section reinforcing the front of the rail starts from nothing just inside of the uprights and swells to its maximum dimension, horizontally, in the middle of the machine. With this latter construction the entire torsional strain is transferred to each end of the rail through the comparatively light section, while with the extended back the strain on the light section of rail which passes in front of the uprights is relieved considerably by the clamps securely holding the back edge of the crossrail to the uprights.

Fig. 2 shows a large planing machine erected in the shops of one of the large electrical manufacturing companies. A conception of the proportions of this machine can be formed by comparison with its surroundings and with the height of the workman standing on the table. It is made with a table to plane work 36 ft. long. The overall length of the bed is 65 ft. 8 in. The clearance between the housings is 16 ft. 2 in. and the distance from the table to the underside of the crossrail is 13 ft. 4 in. These figures fix the limits for the size of the work that can be handled on it. The bed weighs 123,400 lb. and the table (38 ft. 6 in. long) 151,000 lb., the total weight of the machine being 412,340 lb.

On a machine of this type the outstanding considerations are as follows: It must be so constructed that it will produce flat and parallel surfaces throughout its complete working range. The parallel accuracy is of course controlled to a great extent by the foundations upon which the machine is installed. Settling of foundations can very easily throw a long bed out of line to an extent that will affect this requirement. Where accurate work is required from a planer, the bed should be tested, for level and straight, at short periods, for some time after installation and thereafter at such periods as may be indicated by the character of the output of the machine. In the builder's shop, however, the machine must be set up so that it will plane flat and parallel within extreme limits of a few thousandths of an inch.

The same problem confronts the designer in this case as in that of the boring mill, namely, the design and construction of the rail on which the planing toolholders are carried. As the planers so far built have not reached the width between the uprights required on boring mills, it has not been necessary to make the crossrails of as heavy a section as on the boring mills; but the same considerations, on a reduced scale, indicate the desirability of the use of the arched girder to eliminate the initial deflection of the rail.

Planer Reciprocating Mechanism

Another feature of great importance is the question of adequate mechanism whereby the heavy table, and heavy work carried on the table, may be reciprocated at desirable speeds. The planing machine cuts in one direction. The two-direction-cutting planer came into and passed out of existence some years ago. The return stroke of the platen and work is therefore lost time, and must be reduced to the smallest possible amount in efficient production. The time required at each end of the stroke during which the planer must be slowed down, stopped, and reversed is another important item in production. A planer of this size can conceivably be used on a job requiring a stroke of 2 ft. in which the table loading is very high, say, 90 to 100 tons. Imagine the problem of reciprocating this heavy load, in addition to that of the table, backward and forward, cutting on one stroke and coming back to the cutting position on the other. The shock upon the mechanism and upon the fastenings holding the work to the table is obviously one that must be handled carefully. Up to the present time the table-driving mechanism of a planer has assumed two distinct forms. The one more commonly

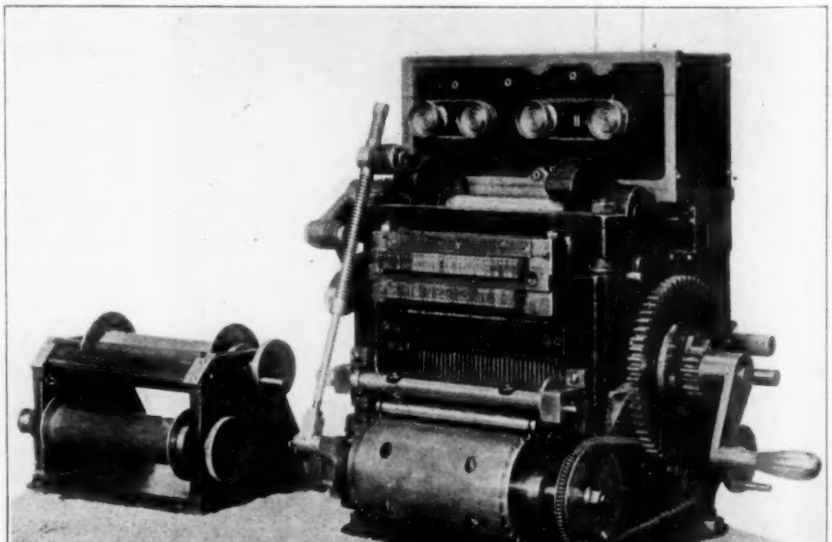

Fig. 7 Mechanism for Punching Paper Templates Used in Automatic Punching Machine Shown in Fig. 6. The Punched Template is Checked on the Rolls Shown at the Left

used is the obvious one and therefore the oldest, wherein the driving mechanism consists of a train of spur gears, reducing the speed from the power means to the proper speed of the work table or platen. These gears are sometimes modified to herringbone types. The power is delivered in all cases of recognized design through a rack bolted or clamped to the underside of the platen. The other type of drive, a recognized form for more than fifty years, is that known to the trade as the "Sellers" drive. The pinion which engages the rack is a helical pinion, commonly called "spiral pinion," the teeth being wrapped around the pinion in the form of a helix. The pinion is mounted on a shaft which extends diagonally through the bed, emerging at the back of the right-hand upright, and to which the necessary reducing gears are applied. This latter form of drive, while more difficult to produce, has an advantage in the fewer reductions necessary to translate the speed from the driving means to the table. This is for the reason that a spiral pinion can be made with four or five teeth, and of a diameter large enough to key upon the driving shaft. In this drive, when properly designed, there are at all times portions of four teeth engaged with the rack. In the case of the spur-geared drive it is difficult, even with a large driving gear with a great number of teeth, to get more than one tooth in positive action at all times. It can be readily seen that with a five-tooth pinion driving into the rack, as compared to at least a twelve-tooth pinion driving through an idler into the rack, a gearing reduction equivalent to more than two to one is introduced.

Coming back to the shock of reversal of the planer platen, the designer using a drive in which the pressure is distributed over four teeth of the rack has an easier problem to solve than one in which one tooth only is counted on to take the shock.

In the planing machine in particular, the designers and builders have to thank the electrical engineers for the best solution, so far, of the problem of reversing driving means. A reversing motor having direct connection to the driving gear of the planer has been developed in capacities up to 75 hp. and both it and the control have been found to stand up successfully under the strain of continuous reversal.

The electric control of the variable-speed d.c. motor has had considerable influence upon the design of machine tools in the last twenty-five years. The facility with which speeds can be changed within the range of the motor, in the case of a drive; and the ease with which one can use remote control on motors for the adjustments, has eliminated many mechanical complications and made possible greater simplicity of design.

Pneumatic Clamping Device

Compressed air is used for some of the operations which in the smaller and earlier machine tools were performed manually. A notable instance of this is found in these same large boring and turning mills and planers. Rigidity is essential to the satisfactory performance in cutting and the crossrail of a mill or planer must be firmly clamped in position during the cutting operation. For this purpose adequate clamps are provided to the inside and outside of the housings. When it is desired to change the distance between the platen or work table and the crossrail, these clamping bolts must be released, and tightened again after setting. On a high, wide machine these bolts are in inconvenient positions for the operator to reach. To obviate this, pneumatic clamping devices are arranged so that the operator from a position on the floor can, by moving a small air valve, control either of the required operations. Fig. 3 shows one of these air-operated clamps at the back edge of an extended-back crossrail. Similar devices are located at the three other clamping points. A cylinder operates two clamps, each one acting to resist the pressure required to set up or unlock its mate. The pneumatic cylinder functions through mechanism which, when clamped, stays in that condition until positively unlocked. It is unnecessary to maintain pressure in the cylinder during the periods between adjustments. The mechanism in the pneumatic clamping devices consists in some cases of screws operating through levers with knife edges to multiply the clamping pressure, the screws being operated by levers. The clamping cylinder is connected to one of the levers, and the end of the piston connected to the mating lever. The cylinder is suspended between the two and acts as an equalizing as well as a pressure means.

Boring mills and planers have been selected from which to draw the few examples given, as being the two outstanding largest types of surface-finishing machine tools constructed up to the present time. Indications are that still larger planers and boring mills will be built in the near future than have been built before.

Automatic Punching Machine

There are other large machine tools in which developments of equal importance with those mentioned have taken place, and their design and construction offer many interesting problems. In figure 6 one such tool is illustrated. This machine is classed as a large machine tool, was designed and built by machine tool builders, yet it might be just as appropriately classified under the title of "automatic machinery." It has been distinctively named by its users the "pianola punch," this name being derived from the fact that its operation is entirely controlled by a punched paper template on a roll, similar to those used in player pianos. Compressed air controlled by the holes in the paper template, operates all of the selective mechanisms.

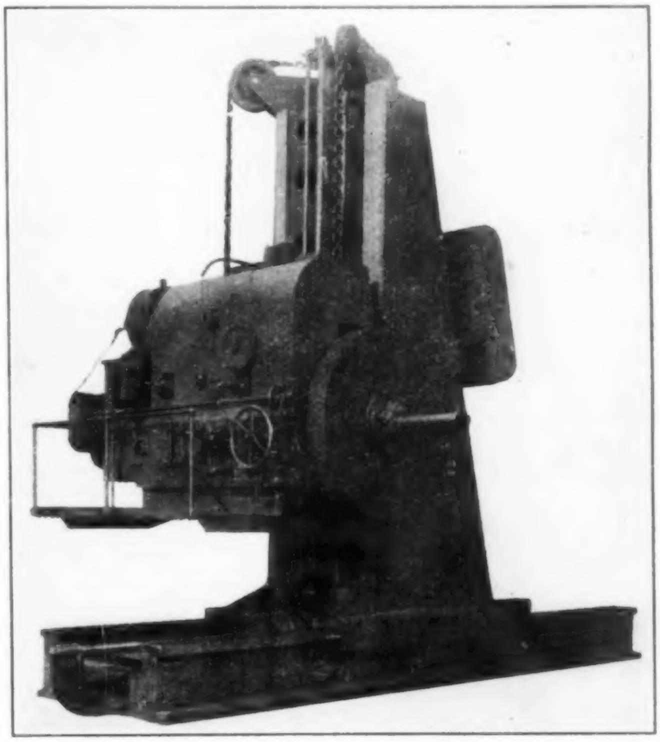

FIG. 8 FLOOR BORING, DRILLING, AND MILLING MACHINE

(A large tool capable of handling the work of a boring mill or planer and often used for boring and surfacing castings for which a planer or boring mill of sufficient size is not available.)

The machine is of the lever type. The punches are carried on a crossbeam and arranged so that their cross-spacing may be adjusted to suit required conditions. Each punch holder is complete with a pneumatic gag-block for throwing its individual punch into or out of engagement. Spacing mechanism is provided for feeding the work through the machine in any multiple of $^1/_{16}$ in. The template which controls all of the functions of the machine is punched and proof read in some office department, brought to the machine, the work clamped in position and machine set in motion. No further attention is needed until the operation of punching the plate or angles, or both, as the case may be, is complete. Then the punch will automatically stop. Innumerable reproductions may be made from the original template, and made with great accuracy. As many as eight plates, separately punched, have been piled one upon the other and the holes registered so closely that rivet-size iron could be put through all of the plates, in any hole. The mechanism for punching and checking the paper templates is shown in Fig. 7.

The discussion on Large Machine Tools might be made to fill up almost an unlimited space; but it would seem as if the foregoing will suffice to broadly outline the general character of the work of designing and constructing large machine tools.

The Efficiency of the Scotch Marine Boiler

Results of Coal- and Oil-Burning Tests Made by the U. S. Shipping Board and the Bureau of Mines Showing the High Combined Efficiencies of Boiler, Superheater and Air Heater That Are Obtainable with Careful Operation

By C. J. JEFFERSON,[1] NEW YORK, N. Y.

THE following brief account of the tests run on a Scotch marine boiler by the U. S. Shipping Board in conjunction with the Bureau of Mines has been written with the idea of presenting to the marine world a few facts and figures in regard to the possible efficiencies and capacities that may be obtained with the Scotch marine boiler, provided it is operated with the same care in supervision as that given to the average efficient stationary plant.

From the analysis of the logs of approximately 250 vessels, it is apparent that the average merchant marine cargo carrier develops approximately 60 to 65 per cent efficiency in her boiler plant when oil-fired and that these values are from 55 to 60 per cent for coal-fired boilers. This same condition holds true for the average stationary plant of less than 1000 boiler hp., as has been shown by several analyses made at various times of boiler tests conducted on this type of plant.

The time has come, however, when better results must be obtained. The Diesel engine is rapidly entering the marine field and the steam-driven vessel must develop its maximum efficiency if it hopes to be able to sail in competition with the motorship, even after making all allowances for the relative difference in the first cost for installation.

Moreover, the merchant marine of the United States has established a higher standard of living for its operating personnel than its competitors. This higher standard means increased cost of operation, and this increased cost must be met by increased efficiency of performance.

The first essential in obtaining higher efficiencies is to educate the operating personnel to the point where they appreciate what higher efficiencies mean, how to use the instruments necessary for determining the efficiency of combustion, and how to correct faulty combustion and thereby build up boiler efficiency.

For this purpose the U. S. Navy has now opened a school at the Philadelphia Navy Yard for the benefit of the U. S. Merchant Marine, where a short and intensive course is given in the theory and practice of fuel-oil burning. Candidates for this school must be American citizens holding merchant marine licenses as second assistant or higher.

The tests which are outlined below were conducted by the U. S. Shipping Board in order that reliable and accurate data might be obtained as to the possible efficiency and capacity of the Scotch marine boiler, which is one of the oldest types of boilers in service and which has been accepted as a good, reliable servant without due regard being paid to its possible efficiency.

The Bureau of Mines was especially interested in making the thermodynamic survey of the boiler by means of thermocouple readings, gas analyses, etc., to determine the effects of combustion by the water-cooled furnace. Its findings are now being prepared by Mr. Mumford of the Bureau of Mines and his associates who worked with him and the author in carrying out these tests.

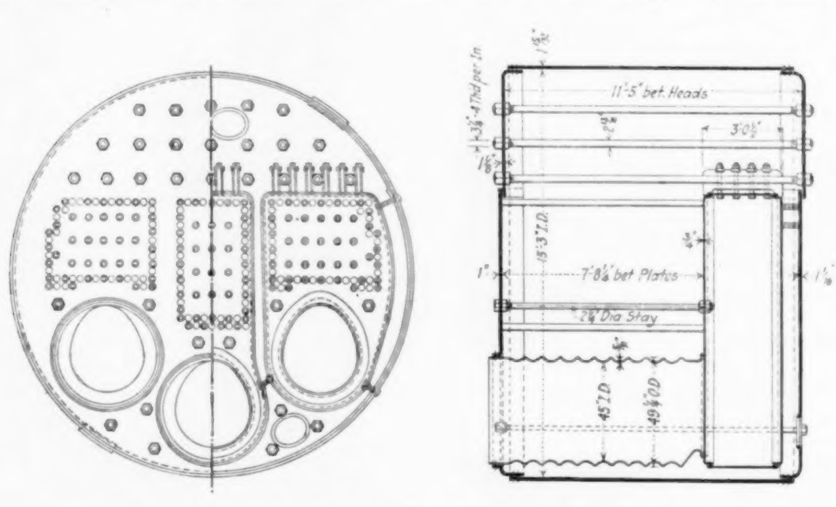

Fig. 1 General Arrangement of Scotch Marine Boiler Tested

The boiler selected for the test purposes was a single-ended, three-furnace, separate-combustion-chamber type, having 2777 sq. ft. of heating surface with coal fire and 3022 sq. ft. with oil fire. The boiler was fitted with a Foster waste-heat superheater having 774 sq. ft. of heating surface and which was placed within the gas pass above the smokebox. Above this superheater was an air heater having 1220 sq. ft. of heating surface which heated the air supply to the Howden fronts when running forced draft. Fig. 1 shows the general arrangement of the boiler and Fig. 2 that of the superheater.

Tests were divided into four distinct groups, namely, hand-fired coal, pulverized coal, forced-draft oil fire, and induced-draft oil fire. The pulverized-coal experiments were conducted to the point where it was demonstrated that this type of firing is not feasible for marine service of the Scotch boiler, as the limited size and water-cooled feature of the furnaces and combustion chambers prevented operating at ratings which were sufficiently high to meet the demand made by the average marine boiler plant.

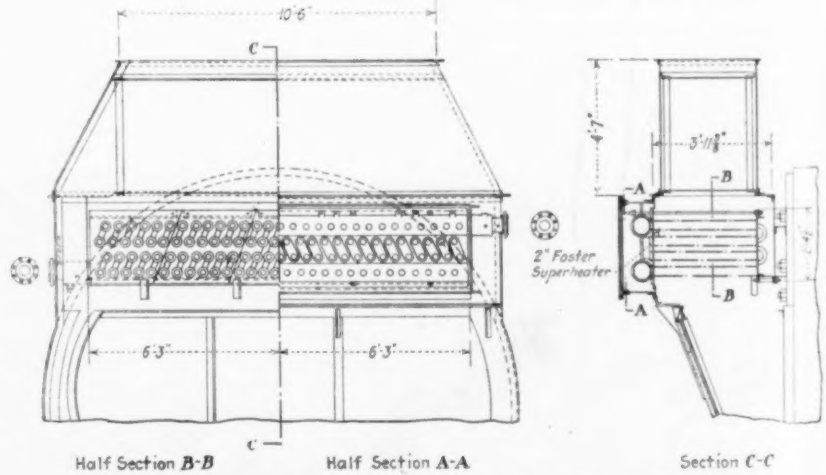

Fig. 2 Superheater Used with Scotch Marine Boiler Tested

Fig. 3 shows the boiler fitted for the hand-fired coal-burning test. In this figure the smokebox, ashpit, and fire doors are shown wide open as it was taken immediately after thorough cleaning of the

[1] Head of Fuel Conservation Section, United States Shipping Board.

Presented at a meeting of the Metropolitan Section of The American Society of Mechanical Engineers, New York, November 14, 1922. Slightly abridged.

TABLE 1 HAND-FIRED COAL TESTS OF SCOTCH MARINE BOILER

Boiler: 3-furnace, with separate combustion chambers; heating surface, 2777 sq. ft. grate surface, 61.8 sq. ft.; area through tubes, 11.94 sq. ft.; combustion chamber volume, 289.4 cu. ft.; heating surface of air heater, 1220 sq. ft.; superheater heating surface, 774 sq. ft.; retarders in tubes; bridge wall, C.I.V.; Coal used, Georges Creek run of mine.

Test number	1-1-1-10	1-1-1-9	1-1-1-8	1-1-1-11
Duration, hr	12.0	10.0	8.083	6.089
Total fuel fired, lb	9,696	12,000	12,521	11,619
Fuel as fired, lb. per sq. ft. of grate surface per hr	13.07	19.4	25.1	30.9
Proximate analysis of fuel:				
Moisture per cent	2.15	2.27	1.97	2.17
Fixed carbon, per cent	70.93	69.87	70.00	69.44
Volatile, per cent	18.84	19.07	19.52	19.14
Ash, per cent	8.08	8.79	8.51	9.25
Heating value, B.t.u. per lb	13,329	13,779	13,858	13,683
Ultimate analysis of fuel:				
Hydrogen, per cent	4.61	4.58	4.58	4.56
Carbon, per cent	80.08	79.34	79.73	79.01
Nitrogen, per cent	1.91	1.81	1.91	1.89
Oxygen, per cent	4.43	4.52	4.27	4.42
Sulphur, per cent	0.89	0.88	0.95	0.87
Ash and refuse:				
Total weight, lb	1200	1340	1309	1125
Per cent referred to dry coal	12.66	11.40	10.60	9.90
Heating value, B.t.u. per lb	3110	4540	4120	4938
Flue-gas analysis:				
CO_2, per cent	11.2	12.2	12.3	11.8
O_2, per cent	6.8	6.6	6.4	7.3
CO, per cent	0.07	0.2	0.3	0.02
N_2, per cent	81.88	81.0	81.0	80.88
Boiler pressure, gage, lb. per sq. in.	163	157	166	158
Superheat, deg	1	14	23	38
Moisture in steam leaving boiler, per cent	0.58	0.50	0.55	0.57
Drafts and air pressures (inches of water):				
Below grates	+ 0.19	+ 0.54	+ 0.605	+ 0.900
Furnace	+ 0.02	+ 0.16	+ 0.283	+ 0.380
Below air heater	— 0.08	— 0.058	— 0.065	— 0.090
Base of stack	— 0.12	— 0.122	— 0.092	— 0.130
Gas temperatures (deg. fahr.):				
Leaving boiler	437	503	537	606
Leaving superheater	372	447	472	519
Leaving air heater	277	336	361	396
Total water fed to boiler, lb	100,560	124,670	122,328	116,988
Feedwater temperature, deg. fahr	212	204	193	208
Equivalent evaporated steam from boiler, lb. per hr. per sq. ft. of heating surface	3.14	4.71	5.78	7.4
Actual evaporation per lb. of coal fired	10.37	10.40	9.76	10.07
Efficiency of boiler without superheater, per cent	75.2	76.7	72.6	74.6
Heat balance:				
Heat absorbed by boiler, per cent	75.8	77.9	73.9	76.8
Loss due to moisture in fuel, per cent	0.2	0.2	0.2	0.2
Loss due to burning hydrogen in fuel, per cent	3.4	3.5	3.5	3.6
Loss due to heat carried away in dry gas, per cent	6.2	7.1	7.4	9.3
Loss due to CO, per cent	0.4	0.9	1.3	0.[illegible]
Loss in combustion—ash and refuse, per cent	2.5	3.3	2.8	3.1
Loss unaccounted for, per cent	11.5	7.1	10.9	6.9
Heat in fuel as fired, per cent	100	100	100	100

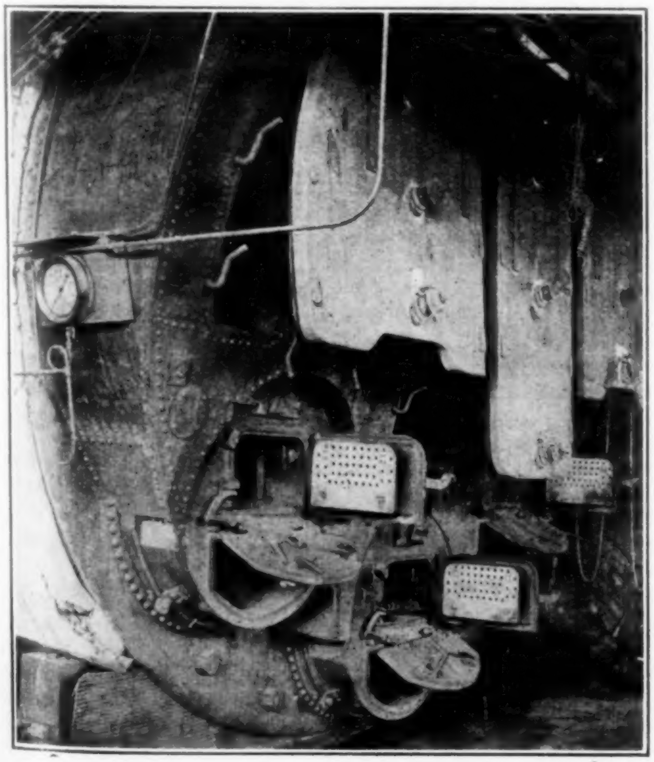

FIG. 3 SCOTCH MARINE BOILER AS FITTED FOR HAND-FIRED COAL-BURNING TEST

fire side of the boiler in preparation for the test. Four of the coal-burning tests which show typical results are given in Table 1. These tests are selected as representing results covering a combustion range of from 13.07 lb. to 30.9 lb. of coal per hour per square foot of grate surface. It will be noted that the plant efficiency including boiler, superheater, and air heater ranges from 73.9 to 77.9 per cent. All of these tests were run under forced-draft operating conditions.

The forced-draft oil-burning tests were conducted using Dahl,

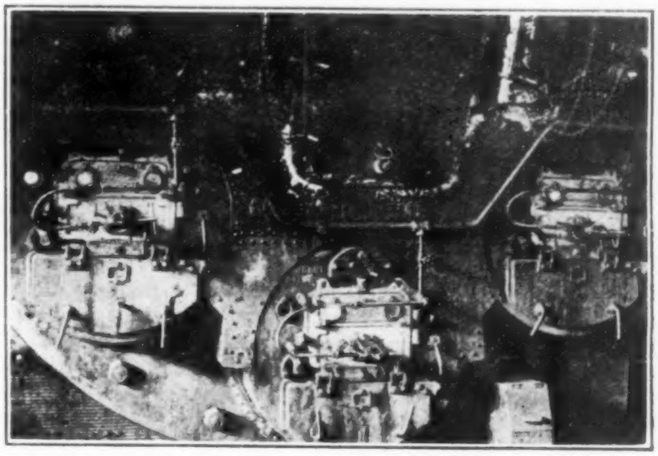

FIG. 4 SCOTCH MARINE BOILER AS FITTED WITH BURNERS INSTALLED IN HOWDEN FRONT FOR FORCED-DRAFT RUNS

Schutte & Koerting, Coen, Todd, and Bethlehem Shipbuilding Corporation[4] burners, all of these except the last named being used

TABLE 2 FUEL-OIL TESTS OF SCOTCH MARINE BOILER—HOWDEN FRONT FORCED DRAFT

Boiler: 3-furnace, with separate combustion chambers; heating surface, 3022.4 sq. ft.; heating surface of air heater, 1220 sq. ft.; superheater heating surface, 774 sq. ft.; retarders in tubes.

Test number	26	36	43
Duration, hr	8.03	2.888	3.026
Combustion space, cu. ft	561	545	588
Total fuel fired, lb	6704	3794	6091
Fuel per burner per hr., lb	278	438	671
Fuel per hr. per sq. ft. of heating surface, lb.	0.276	0.435	0.666
Analysis of oil as fired:			
Moisture, per cent	0.9	0.3	0.3
Sediment, per cent	trace	trace	trace
Hydrogen, per cent	11.9	11.72	11.6
Carbon, per cent	83.3	83.83	84.5
Sulphur, per cent	3.9	4.15	3.6
Gravity, deg. B	15.4	15.4	15.4
Specific gravity	0.963	0.963	0.963
Flash point, deg. fahr	166	166	166
Burning point, deg. fahr	234	234	234
Heating value, B.t.u. per lb	18,193	18,324	18,234
Flue-gas analysis:			
CO_2, per cent	10.8	12.3	12.6
O_2, per cent	6.3	4.5	3.7
CO, per cent	0.03	0	0
N_2, per cent	82.87	83.2	83.7
Weight of gas per lb. of fuel, lb	19.37	16.9	16.83
Boiler pressure, gage, lb. per sq. in	170	172	178
Superheat, deg	24	32	45
Moisture in steam leaving boiler, per cent	0.65	0.62	0.62
Drafts and air pressures (inches of water):			
Furnace	+ 0.29	+ 0.19	+ 1.28
Below superheater	— 0.08	— 0.09	— 0.05
Below air heater	— 0.11	— 0.11	— 0.09
Base of stack	— 0.14	— 0.14	— 0.11
Gas temperatures (deg. fahr.):			
Leaving boiler	533	597	665
Leaving superheater	470	512	571
Leaving air heater	341	385	428
Total water fed to boiler, lb	97,635	56,574	88,130
Feedwater temperature, deg. fahr	188	218	206
Equivalent evaporated steam from boiler, lb. per hr. per sq. ft. of heating surface	4.29	6.72	10.10
Actual evaporation, lb. per lb. of oil fired	14.57	14.91	14.47
Efficiency of boiler without superheater, per cent	82.80	81.80	80.73
Heat balance:			
Heat absorbed by boiler and superheater, per cent	84.6	83.8	83.5
Loss due to moisture in fuel, per cent	0	0	0
Loss due to burning hydrogen in fuel, per cent	6.0	5.6	6.0
Loss due to heat carried away in dry gas, per cent	3.8	3.9	4.4
Loss due to CO, per cent	0.2	0	0
Loss in unconsumed oil and unaccounted for, per cent	5.4	6.7	6.1
Heat in fuel as fired, per cent	100	100	100

in conjunction with the Howden combined oil- and coal-burning front.

Fig. 4 shows the boiler fitted with burners installed in the Howden

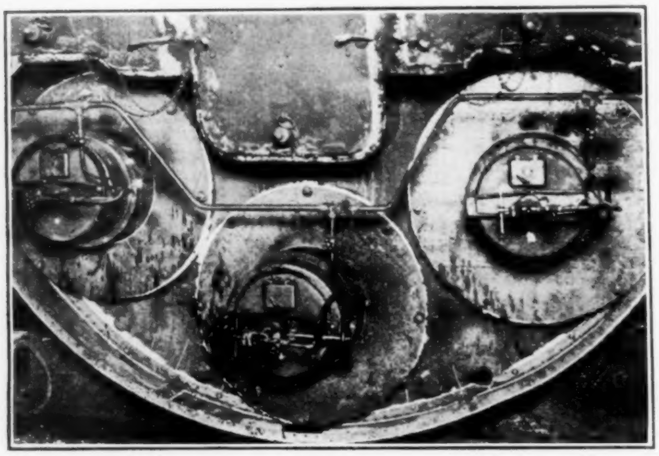

Fig. 5 Scotch Marine Boiler as Fitted with Bethlehem Burners for Induced-Draft Tests

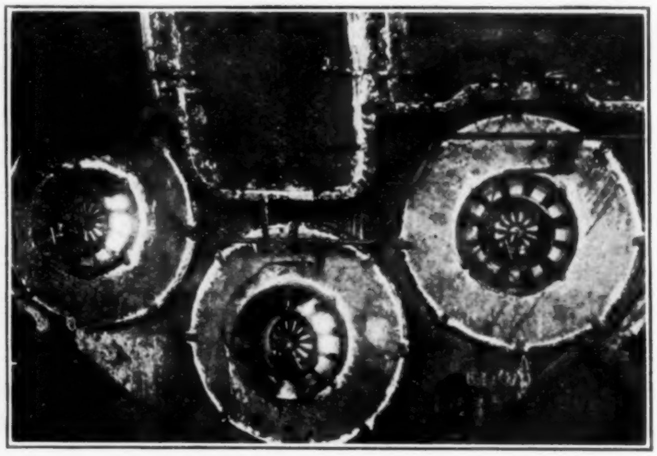

Fig. 6 Scotch Marine Boiler as Fitted with Enco Burners for Induced-Draft Tests

front for forced-draft runs. Table 2 gives the results for three representative tests of this series covering a range of from 0.276 to 0.666 lb. of oil per hour per square foot of heating surface. It will be noted that the combined boiler, superheater and air heater efficiency ranges from 83.5 to 84.6 per cent. Fig. 7 gives the average efficiency values covering the entire series of forced-draft oil-burning tests.

The induced-draft tests were conducted with the Bethlehem Shipbuilding Corporation burner shown in Fig. 5, as well as with the Schutte & Koerting types "L" (modified) and "N," the Coen, the Todd, and the Enco burners. This latter burner, Fig. 6, was originated at the Philadelphia Navy Yard Fuel Oil Test Plant, and its adaptation for use in Scotch boiler service was developed during these tests.

TABLE 3 FUEL-OIL TESTS OF SCOTCH MARINE BOILER—NATURAL DRAFT REGISTERS

(Boiler same as used in tests of Table 2)

Test number	97	94	67	79
Duration, hr.	3.00	2.498	3.203	2.981
Combustion space, cu. ft.	560	560	561	564
Total fuel fired, lb.	2111	2868	4408	4625
Fuel per burner per hr., lb.	235	383	459	517
Fuel per hr. per sq. ft. of heating surface, lb.	0.233	0.380	0.455	0.513
Analysis of oil as fired:				
Moisture, per cent	0	0.1	0.1	0
Sediment, per cent	trace	trace	trace	trace
Hydrogen, per cent	11.37	11.25	11.34	11.78
Carbon, per cent	84.08	83.82	83.60	83.98
Sulphur, per cent	4.0	3.9	3.5	3.8
Gravity, deg. B.	15.4	15.4	15.4	15.4
Specific gravity	0.963	0.963	0.963	0.963
Flash point, deg. fahr.	166	166	166	166
Burning point, deg. fahr.	254	254	254	254
Heating value, B.t.u. per lb.	18,382	18,418	18,248	18,459
Flue-gas analysis:				
CO_2, per cent	12.8	13.06	12.2	11.98
O_2, per cent	3.3	3.0	4.4	4.4
CO, per cent	0.03	0.17	0.01	0.06
N_2, per cent	84.87	83.77	83.39	83.56
Weight of gas per lb. of fuel, lb.	16.48	15.95	17.19	17.49
Boiler pressure, gage, lb. per sq. in.	172	174	172	168
Superheat, deg.	0	14	28	33
Moisture in steam leaving boiler, per cent	0.62	0.64	0.62	0.47
Drafts and air pressures (inches of water):				
Furnace	−0.14	−0.22	−0.31	−0.64
Below superheater	−0.15	−0.25	−0.48	−0.78
Base of stack	−0.15	−0.31	−0.57	−0.84
Gas temperatures (deg. fahr.):				
Leaving boiler	448	525	589	548
Leaving superheater	420	473	503	490
Total water fed to boiler, lb.	32,276	41,282	61,813	66,348
Feedwater temperature, deg. fahr.	232	212	191	212
Equivalent evaporated steam from boiler, lb. per hr. per sq. ft. of heating surface	3.63	5.69	6.79	7.68
Actual evaporation, lb. per lb. of oil fired	15.29	14.39	14.02	14.35
Efficiency of boiler without superheater, per cent	82.35	78.98	79.29	78.70
Smoke, Ringlemann scale	1.5	1.75	1.5	2.5
Heat balance:				
Heat absorbed by boiler and superheater, per cent	82.9	80.2	81.1	80.6
Loss due to moisture in fuel, per cent	0	0	0	0
Loss due to burning hydrogen in fuel, per cent	5.7	5.7	6.0	6.1
Loss due to heat carried away in dry gas, per cent	7.1	8.4	9.6	9.4
Loss due to CO, per cent	0.1	0.6	0	0.2
Loss in unconsumed oil and unaccounted for, per cent	4.2	5.1	3.3	3.7
Heat in fuel as fired, per cent	100	100	100	100

The ratings obtained in these induced-draft tests were limited by the amount of available draft as the height of stack above the center of furnace was only 40 ft., and this draft was augmented by a ½-in. steam nozzle. The combined effect of the stack and nozzle produced an 0.9-in. draft at the base of the stack. Table 3 gives the data of four representative tests covering a range of from 0.233 to 0.513 lb. of oil per hour per square foot of heating surface and showing efficiencies ranging from 80.6 to 82.9 per cent. Fig. 8 gives the average efficiency values obtained from all of the tests of this class.

In comparing Figs. 7 and 8, the fact must be kept in mind that the induced-draft tests did not have the help of the air heater in building up furnace efficiency.

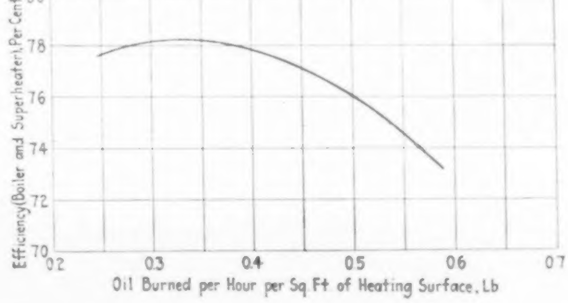

Fig. 7 Average Combined Efficiency Values Obtained in the Forced-Draft Oil-Burning Tests

(1002.46 × lb. oil per sq. ft. of heating surface = oil per burner.)

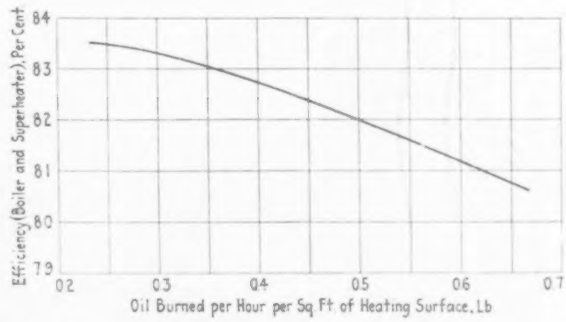

Fig. 8 Average Combined Efficiency Values Obtained in the Induced-Draft Tests

(1002.46 × lb. oil per sq. ft. of heating surface = oil per burner.)

From the foregoing data it will be seen that the Scotch marine boiler is capable of efficiencies considerably in excess of the average operating values obtained, and it is, therefore, a matter of simpel calculation to show what considerable fuel savings are possible when reasonable supervision is shown on the part of the operating force.

Effect of Pulsations on Flow of Gases

By HORACE JUDD,[1] and DONAL B. PHELEY,[2] COLUMBUS, OHIO

The movement of gases and liquids when calculated by the existing hydraulic formulas presupposes a steady or continuous flow of the fluid. Anything which causes this flow to proceed in puffs, waves, or pulsations will result, by the action of metering devices, in errors often of great magnitude which generally do not admit of any adjustment, or of any definite knowledge of the amount of the error.

The present paper discusses work undertaken under the joint direction of the Engineering Experiment Station of the Ohio State University and the Research Sub-Committee on Fluid Meters of The American Society of Mechanical Engineers, which had for its object (1) the study of the nature of the pulsation and (2) the discovery of some practical means of reducing or eliminating the pulsation or of compensating for its effects on the devices used for measuring fluid flow. The investigation was confined to the venturi meter, the orifice, the flange nozzle meter, and the pitot meter, using air flow from a small compressor discharging into a 3-in. line. It is believed, however, that the basic principles established by the experiments are fundamental for pulsating-flow conditions for gas, steam, and water as well as for air, and also for other sizes and kinds of installations.

ONE of the most disturbing factors encountered in recent years in the metering of air, gas, steam, and water, especially in connection with all forms of power engineering, has been that due to turbulent or pulsating flow. This has not been confined to any one class or type of meter, but is present to a more or less degree with all forms of metering devices.

The measurement of gases and liquids when calculated by the existing hydraulic formulas presupposes a steady or continuous flow of the fluid. Anything which causes this flow to proceed in puffs, waves, or pulsations will result, by the action of metering devices, in errors often of great magnitude which generally do not admit of any adjustment, or of any definite knowledge of the amount of the error.

The work described in the present paper was undertaken under the joint direction of the Engineering Experiment Station of The Ohio State University, and the Research Sub-Committee on Fluid Meters of The American Society of Mechanical Engineers. A sub-committee of the Fluid Meters Committee consisting of A. R. Dodge, H. N. Packard, and H. Judd was selected to take direct charge of the research work.

Purpose of the Investigation

The object of the investigation as outlined by the sub-committee in direct charge was twofold:

a To study the nature of the pulsation

b To discover some practical means of reducing or eliminating the pulsation, or of compensating for its effects on the devices used for measuring fluid flow.

For convenience, flow meters have been classified in two main divisions which may be called (1) positive meters, and (2) inferential meters. The domestic gas, or water, meter is an example of the first class. It is a displacement meter in which an actual volume of gas is introduced into a container of known size, and the quantity thus measured is registered on the meter.

In commercial installations of even moderate size, inferential meters are used almost entirely. In meters of this class some function of the quantity of fluid passing a given cross-section of pipe is measured and from this observation the actual flow is deduced or "inferred." This method can be made to give accurate results under steady-flow conditions, but when the flow is pulsating the accuracy of the measurement is seriously affected, if, indeed, not entirely destroyed.

Three general cases may be mentioned where this problem is of great importance: (1) The measurement of natural gas, both entering and leaving a compressor station where reciprocating compressors are used; (2) the measurement of air both entering and leaving large reciprocating air compressors, or blowing engines; and (3) the measurement of steam supplied to reciprocating steam engines. The steam flow is pulsating in character because the engine cuts off the steam supply during a considerable part of each stroke. In each of these cases the flow of the fluid has a regular, comparatively rapid, rhythmical pulsation, which occasions serious errors in measurement, especially where the measuring element is of the inferential type.

Similar pulsating conditions are present in water flow where reciprocating pumps are used. The problem, however, is more easily solved by the proper use of air chambers and surge tanks. Water hammer in pipe lines from whatever cause bears a striking similarity to the pulsating effect produced by an air-compressor valve.

The authors have confined their investigations to inferential meters. These meter elements as selected are the venturi meter, the orifice meter, the flange nozzle meter, and the pitot meter. Furthermore, they have been limited to air flow from a small compressor discharging into a 3-in. line; and hence their findings, strictly speaking, would be applicable only to installations of similar character. However, it would seem highly probable that the basic principles established by these experiments would be fundamental for pulsating-flow conditions for gas, steam, and water as well as for air, and also for other sizes and kinds of installations.

Equipment Employed in the Investigation

The experimental work was carried on in the Mechanical Engineering Laboratories of the Ohio State University, and was begun in May, 1920. The essential elements for carrying on the project were: (1) a disturbing element to produce the pulsating flow; (2) a quieting element, or elements, to eliminate or modify the pulsations; and (3) a measuring element, to indicate constant flow conditions and also to indicate the effect of the pulsating flow.

Disturbing Elements. Fig. 1 shows the general layout of the apparatus, at the extreme left hand of which is located the air compressor C, a 9-in. by 9-in. single-stage, single-acting, gas engine-driven machine running at 293 r.p.m. This supplied air to a line about 120 ft. in total length of which 50 ft. was made up of $2^1/_2$-in. pipe containing several short lengths and fittings. The remaining 70 ft. comprised the 3-in. test line of straight continuous length. This test line was at first made up of standard 3-in. black pipe of commercial quality; later 24 ft. of 3-in. brass pipe was substituted for that portion of the test line preceding the meter and extending 3 ft. below the meter section. (See Fig. 1, B.)

This air supply with its full pulsating effect could be admitted directly to the test line or could be first discharged through a large tank before entering the test line. A second disturbing element for producing pulsations artificially is shown at I, Fig. 1, the butterfly-valve interrupter. This butterfly valve could be driven at speeds ranging from 180 r.p.m. to 800 r.p.m., producing thereby a variation in the number of pulsations per second.

Quieting Elements. Considerable study was made of this essential feature and many trials were made to satisfy the authors that they were securing pulsationless flow where and when needed. The tank T, Fig. 1, was used to quiet the pulsation before the meter station, M, was reached. This was a 48-in. vertical tank of 200 cu. ft. capacity. Air could be admitted either at the bottom or at the top and released from the tank through the internal pipe which reached nearly to the top. This tank when fitted with $1^1/_4$-in. orifices at top and bottom made it possible to get air flow free from pulsations before the test meters were reached.

A second quieting tank was inserted at Q, Fig. 1, at a point 8 ft. below the meter station. This was necessary in order to secure pulsationless flow at the orifice head for the purpose of establishing standard flow conditions. This tank was selected almost by chance and afterward was proved by test to be of sufficient capacity to eliminate practically all of the effect of the pulsating flow, and with

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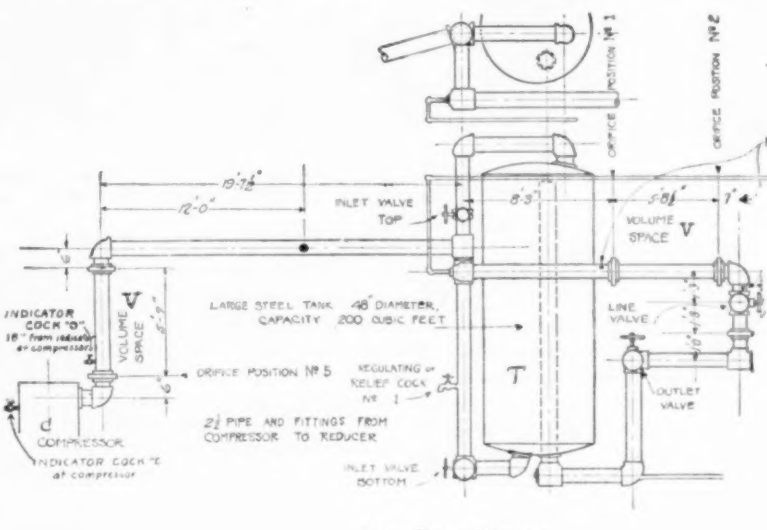
[1] Professor of hydraulic engineering, Ohio State University. Mem. A.S.M.E.

[2] Junior engineer, U. S. Coast and Geodetic Survey.

Contributed by the Research Committee and presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. Abridged. All papers are subject to revision.

the insertion of a $1^1/_4$ in. orifice at the exit from the tank the authors were entirely successful in securing pulsationless-flow conditions.

There are two positions marked, *V*, Fig. 1, one near the compressor and one just beyond the large quieting tank, where volumes of different sizes were inserted for the purpose of studying their quieting effect. Because the term volume seems to apply better the authors have perverted the word "volume" from a term meaning capacity to a special designation, and have used it altogether to denote the various tanks of different dimensions which have been used as quieting elements. Most of these volumes were used in the second volume space, *V*, beyond the large quieting tank.



Left-Hand Portion

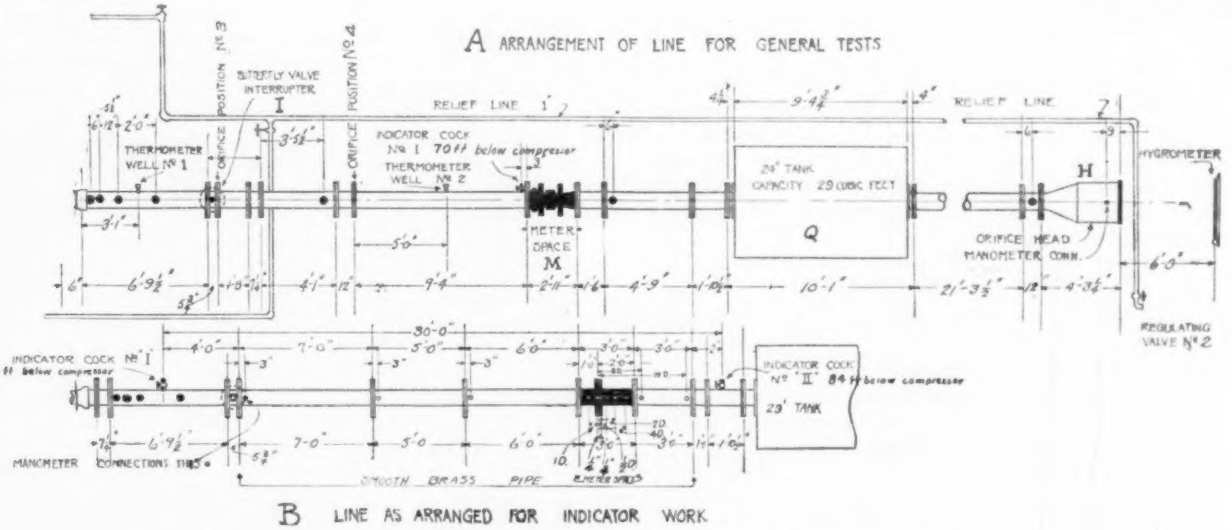


Fig. 1 General Layout of Apparatus, Right-Hand Portion (Left Hand Portion Above)

The line valve near the entrance to the 3-in. test line was used as a quieting element when employed as a throttling device. Both a gate valve and a globe valve and in some cases orifices were thus used as throttling devices.

The combination of throttling with volumes constitute the muffler type of quieting device. Fig. 3, shows at *B* an 8-section pipe-flange muffler which was also inserted in the line at the second volume station. This muffler is located in a by-pass in front of the line which runs directly from the compressor. All the volumes used as quieting devices were placed in direct line, but later it was found that the by-pass position answered just as well for the muffler or the other quieting devices. Fig. 3 shows sectional views of the pipe-flange muffler and also other forms of mufflers, including two funnel mufflers. A form of pulsating bag was also used as a quieting volume and was connected to the compressor line in a way similar to an air chamber on a reciprocating pump. Another device used was a system of revolving fans or baffles.

Measuring Elements. Under this heading will be taken up in order: (1) the orifice-head meter, (2) meter elements used, and (3) manometers used.

Orifice-Head Meter. It was recognized at the outset that one of the indispensable features was an accurate method of measuring the discharge of the pipe, or (its equivalent) an accurate means of indicating the velocity of the air in the pipe. This was effected by means of an orifice head at the end of the pipe line, Fig. 1, *H*. Its outer diameter is 9 in. for a distance of 2 ft., followed by a tapered section to meet the 3-in. line. This is given a taper of 7 deg., and was so chosen as being the limiting angle for preventing as far as possible the swirling and eddying of the air as it passes into the orifice head from the line. Seven holes, reamed to $1^1/_8$ in., were provided in the head plate ($^3/_{16}$ in. thick), although during the tests not more than five were used at one time. In general the capacity of the compressor was reached with four holes open with a standard static discharge head of 0.9 in. of water at the orifice head. The orifice head was calibrated by discharging air through it from the calibrated tank *T* under a constant static head at the orifice head.

The orifice head was also checked against a second orifice to show how uniformly the air was distributed in its cross-section. All possible combinations of the orifices including single orifices and 5-hole orifices with and without the center orifice gave a variation not exceeding one-tenth of one per cent. This established uniform flow conditions in the orifice head regardless of the number of orifices that were open in the head plate.

Meter Elements Used. The point of insertion of the meter elements in the line, Fig. 1, *M*, was about 70 ft. from the compressor, 50 ft. above the orifice head, and 25 ft. from the entrance to the 3-in. test line.

The venturi meter was a standard unit with 3-in. entry and 1-in. throat. It is shown in sectional view at *A* in Fig. 2.

The orifice meter was made up of a flanged section of 3-in. brass pipe of the same length, 35 in., as the venturi section. The orifice flange, *B*, Fig. 2, was placed 12 in. from the upstream end and was counterbored to receive the set of orifice plates and to center them accurately. The downstream side of the hole in each plate was chamfered to $^1/_{32}$ in. in thickness.

The flange nozzle meter was made by inserting in the orifice-meter section a special-shaped rounded-edge orifice with projecting cylindrical end.

Two forms of pitot tips were used. The hatchet-edge static tip

(pitot No. 1) with $^1/_8$-in. side openings is used with an accompanying open-ended impact tip with $^5/_{16}$-in. opening. Both tubes are made of $^1/_4$-in. seamless brass tubing. Also a modified form of pitot tip (pitot No. 2) was used having $^3/_{16}$-in. brass tubing and an impact or leading opening facing the direction of flow and a static or trailing opening directly opposite. The diameter of each opening is $^1/_8$ in.

Manometers Used. As far as possible the simpler forms of manometers were used. The vertical U-tube water manometer, Fig. 2, *D*, was used with the venturi meter, and part of the time with the orifice meter and flange nozzle meter, and also for the static line pressure. Where the readings had to be magnified, use was made of a 6-in. inclined one-leg reservoir oil manometer, 5 to 1 magnification (Fig. 2, *C*). For certain other readings an inclined *U*-tube oil manometer and a vertical *U*-tube two-liquid manometer were used.

A Foxboro differential mercury recording gage, Fig. 2, *E*, was used to make comparisons with the water manometer used with the venturi meter. The flow conditions were maintained and checked at the orifice head by means of an Ellison inclined gage of 1 in. range and 10 to 1 magnification.

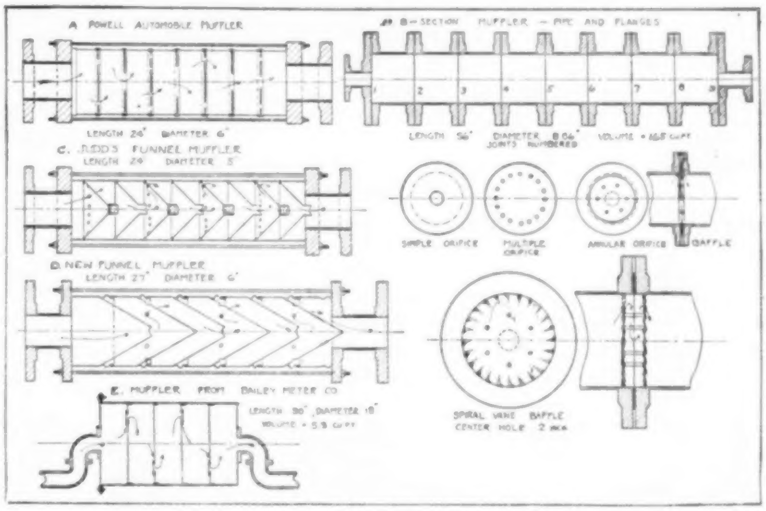


Fig. 3 Muffler Devices

Nature of the Pulsation

The first knowledge of the nature of the pulsation was gained through the use of a "photopulsometer," made and loaned to the authors by Mr. H. N. Packard of the Cutler-Hammer Co., Milwaukee, Wis. In this instrument a pitot tube with one leading and one trailing tip set with the opening in line on a vertical diameter was inserted in the center of the pipe. The leading or impact tip communicated with the under side of a diaphragm chamber. The trailing or static tip led to the upper side of the diaphragm chamber. The mica diaphragm, 0.011 in. thick, would therefore respond to changes in velocity of the air in the line as they occurred. These vibrations were directly transmitted to a mirror hung in jeweled bearings and by means of a beam of light could be thrown on a photographic film giving a diagram proportional to the velocity. By means of a pendulum beating quarter- and half-seconds a chronographic record could also be made as shown on most of the films by the breaks in the diagrams. A great many films were taken in this manner under a number of different running conditions and it proved to be a valuable method for providing a permanent record of the state of the flowing air in the line, either under violent pulsations due to various disturbing factors or for more steady flow due to the effect of certain quieting factors, as well as a record of the state of flow when under steady or pulsationless-flow conditions.

Velocity Diagrams. Figs. 4, 5 and 6, give records of the velocity changes at the center of the pipe line at various points and under various flow conditions. The maximum effect of the pulsation in the open line direct from the compressor is shown in Fig. 4. There is little or no quieting effect due to a length of pipe equal to 183 diameters.

The large tank *T* was tried as a quieting chamber, when connected similar to an air chamber to a pump. However, it was found to be worthless as a quieting device. This substantiates the authors' later experience that, to be effective, tanks, or volumes, should be inserted in the line so that the air may pass through them axially.

One of the methods for eliminating pulsations is throttling by means of some kind of obstruction in the line such as a valve or an orifice. Diagram (*a*), Fig. 5, is taken for maximum pulsations direct from compressor. The diagrams appear to go below the zero pressure line, but when the secondary pulsations due to the natural period of vibration of the diaphragm are considered, it will be seen that the true diagrams approach but do not go below the zero line. Diagram (*b*) shows the quieting effect of an orifice when the pulsation has been reduced to the condition of pulsationless flow.

Effect of Pulsation of the Static Pressure in the Line. On starting the compressor it seemed frequently that the first impulse traveled much faster than the actual velocity of the air. To test this out and also to study the effect of pulsation on the static pressure in the line, two Crosby indicators were attached to the line, one on the compressor cylinder (Fig. 1, *C*), and the other 70 ft. distant (Fig. 1, *A*, No. 1 indicator) near the meter space. Diagrams were taken and two important features were brought out: (1) The pulsation in the pipe produced a much greater pressure effect than that imparted to the velocity of flow; (2) when simultaneous diagrams for a single stroke were recorded it was found that the suction stroke of the compressor corresponded to the pressure stroke in the line. This seemed to indicate that the pulsation required about the time of a compressor revolution to travel a distance of 70 ft. in the pipe. For a speed of 291 r.p.m. this would mean about 0.1 sec. for $^1/_2$ revolution, or a pulsation velocity of 700 ft. per sec.

The Velocity of the Pulsation. The results obtained in determining the velocity of pulsation from the simultaneous sets of indicator diagrams for points located 30 ft. apart established three significant facts:

a That, although the authors' method shows results varying from the maximum to the minimum through a wide range, yet in no case does the velocity of pulsation so determined approach anywhere near the velocity of the flowing air.

b That, for a variation of velocity of flowing air ranging from zero to 27 ft. per sec., the velocity of pulsation was found to be independent of the velocity of the air.

c That the total average for 148 computations gave 1090 ft. per sec. as the velocity of pulsation. The velocity of

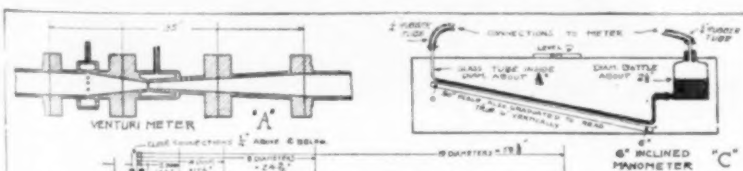


Fig. 2 Section Sketches of Meters, Manometers, and Connections

sound in dry air at 32 deg. fahr. and 29.92 in. barometer is 1083 ft. per sec. The velocity of pulsation in all probability is equal to that of sound in air.

For the average velocity of pulsation equal to 1090 ft. and for 4.88 pulsations per second, the pressure wave length as shown on the diagrams would be 223 ft.

Since the velocity of the pulsation is independent of the velocity of the flowing air and is evidently equal to the velocity of sound in air, it seems quite reasonable to conclude that the pulsation is a pressure change in the form of a wave front resembling a sound wave of low frequency. It seems also highly probable that these pulsations are similar in character to the pulsations set up by water hammer in a pipe line, since they also travel with the velocity of sound in water.

Some additional knowledge of the nature of the pulsation was

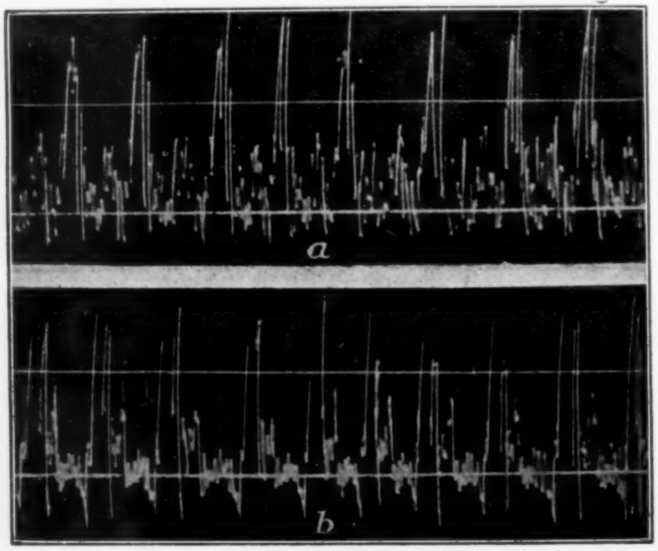


Fig. 4 Showing Maximum Effect of Pulsations at Different Points on the Line—No Quieting Effect Due to Increase in Length of Line

((*a*) Maximum pulsation in line 50 ft. below compressor. (*b*) Do., 111 ft. below compressor.)

gained by noting its effect on manometers. Where a manometer was used to measure a differential head at a meter, the following effects were consistently present:

a For pulsationless flow the reading was very constant, the only variation being a slight long period surge due to appreciable variations in the compressor speed.

b For pulsating flow this surge was magnified greatly.

c For pulsating flow there is also a rapid vibration of the water column corresponding to the pulsations and depending in amplitude upon the local conditions at the manometer.

d The most significant characteristic of the readings for pulsating flow was the large increase over that for pulsationless flow. This increase was present for every type of manometer, meter and gage tried. It varied from a few per cent to several hundred per cent under extreme conditions.

The Elimination of the Pulsation

The problem of the elimination of the pulsation, or of the effects due to the pulsation, suggested two methods of attack: (1) Modification of the existing metering devices so that the recorded flow would be unaffected whether the flow be steady or in pulsations; and (2) the use of devices which would correct or eliminate the pulsations before the flowing fluid reached the meter.

The first of these suggested schemes was taken up to some extent in the study of the modification of manometer connection. The second suggestion, that of pulsation elimination, received the major part of the authors' attention. Of the five quieting devices used, the pulsating bag and the revolving fan operated by the air flow were studied by means of the photopulsometer. The use of throttling devices, the insertion of tanks, volumes or equalizing chambers in the line and a combination of the two devices, forming the so-called "muffler," comprise the remaining three quieting and eliminating devices. These five schemes, it is believed, cover nearly all, if not all, of the practical schemes which might be used for this purpose.

Modification of Manometer Connections at the Meter. The first attempt to reduce the error of pulsation by this means was by throttling the manometer connections. It was found that throttling has no appreciable effect in reducing the error until the opening has been reduced to less than 0.07 in. diameter, and that even for an obstruction so small as nearly to close the opening the percentage of error is not reduced to within practical limits. The surge, or pulsation, of the water column was completely destroyed, so that the effect is quite analogous to that of the steam gage when throttled. This indicates that while throttling a pressure gage does not affect its reading, it has no beneficial effect in reducing the error due to pulsation.

The efficient quieting effect of volumes when used in the test line suggested the possibility that small volumes inserted in the tubes leading to the manometers might serve to reduce the pulsation before the meter was reached.

Several tests were made using the orifice meter and the 6-in. inclined gage with volumes of different sizes in one or both of the manometer connections. The results of these tests show that the use of volumes in the manometer connections does not give so favorable results as the method of throttling. There is only about 15 per cent reduction in the error for the 70 per cent orifice meter.

It was thought, also, that the point of attachment of the manometer connection might have some influence on the error due to pulsation. With the orifice meter comparisons were made with the manometer connected (1) close to the orifice and (2) at a distance of one pipe diameter above the orifice and at points below the orifice ranging from $^{1}/_{2}$ diameter to 19 diameters.

Tests made show that for all orifices including the 80 per cent

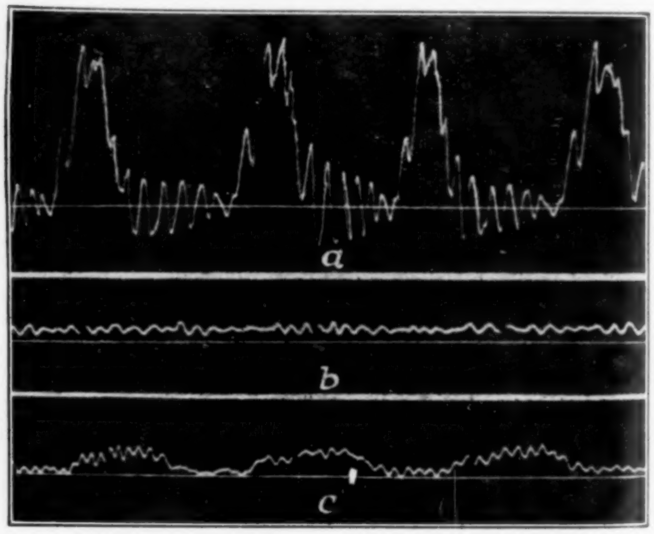


Fig. 5 Showing Quieting Effect of Throttling by Means of an Orifice

((*a*) 80 ft. below compressor, maximum pulsation in line. (*b*) 10 ft. below throttling orifice, pulsation destroyed. (*c*) 4 ft. above interrupter, 3.5 pulsations per sec.)

orifice the error at the standard points of correction is greater than that for points near the orifice which averages 96.3 per cent of the error at the standard points of correction. For points farther distant from the orifice there is a tendency for the error first to increase and then to decrease as the 19 diameters point is approaching, in all covering a total range of 20 per cent error.

Effect of Type of Manometer Used. The error in measuring pulsating flow seemed to depend to some degree upon the type of manometer used to register the head, even if all other conditions of the line and meter were identical.

It may be stated that all manometers properly graduated will give the same head readings for pulsationless flow. But when the flow is pulsating, the ratio of its reading to the true reading will differ somewhat according to the variations mentioned above. A mercury manometer will probably show an error less than that of a water manometer and the latter less than one using mineral oil. A manometer with small tubes is likely to read higher than one

with a larger set of tubes, but this is merely a tendency. If the tubes are too small the capillary effect can be noted; and if they are too large, or if they end in a reservoir, the doubtful effect due to a "volume" will be introduced. The inclined leg of a manometer under some conditions may even cause less "surge" effect. The presence of a check value or damping device between the two manometer legs or chambers, or a float to actuate the recording arm, may reduce the error.

Maximum Percentage of Error Produced by Pulsation. The results obtained show that the less the obstruction to the flow of the air, the greater the percentage of error due to pulsation; also, the greater the restoration of pressure after passing the meter, the greater will be the error. The reason for this relation appears to be that, since the pulsation is a form of pressure energy, that type of meter unit which in itself most completely dissipates the pulsation energy will show the least percentage of error.

Distribution of Pulsation as Shown by Traverse. The pipe was traversed by both types of pitot tubes for maximum pulsation conditions. The results when plotted show by both traverses that error due to the pulsating flow is least at the center of the pipe. There is a slight tendency for the point of minimum error to be located a little to one side of the center of the pipe. It is not known just how much the pitot tubes themselves are influenced by their approach to the wall of the pipe.

Quieting Effect of a Revolving Fan Section. The revolving fan apparently has some merit as a quieting device, but is of questionable practical value.

The Effect of the Pulsating Bag as a Quieting Device is shown by Fig. 6. It is felt that the special design of such a device would be needed to cover the requirement of each individual installation in order to correct or eliminate the pulsating error.

Elimination of Pulsation by Throttling. The experiments carried on with the various devices for eliminating or modifying the pulsation led to the conclusion that the solution of the problem depended entirely on the absorption of the energy of the pulsation propagated as a pressure wave closely resembling a sound wave of low frequency. Whatever the device used, its value in killing the pulsation will be measured by its ability to absorb, or dissipate, this energy of pulsation.

The general effect of throttling is to reduce the error rapidly by means of a pressure drop up to 4 in. of mercury. The use of a greater pressure drop causes the error to be reduced more gradually. In most cases the error is not reducible below 1 to 3 per cent, even with a sacrifice of a drop of 12 in. mercury. The manner of throttling is immaterial whether by gate or globe valve or by an orifice.

Elimination of Pulsation by the Use of Volumes. Tanks, or volume capacities, or "volumes," as the authors have chosen to call them, were used in the line for the purpose of quieting the pulsation. These volumes were inserted in the line so that the direction of flow through them was along the axis of the volume. They were all cylindrical in shape and with the exception of the 8-in. and the 48-in. volumes were made of thin sheet metal, No. 24 gage. It is evident from the results obtained that a volume is also a practical means of eliminating the pulsation. The chief question is, whether in large installations volumes of sufficient size would be of practical use.

Effect of Varying the Shape of Volume was also studied. In a general way a volume is probably more efficient when it is of relatively large diameter.

Elimination of Pulsation by Combining Throttling with Volumes. Since the pulsation could be nearly if not quite eliminated either by the use of throttling devices or by the use of volumes alone, the natural conclusion was that some combination of the two schemes might be discovered which would give satisfactory results without the objectionable large pressure drop or the excessive size of the volume. A series of runs was made, while the various volumes were in the line, where orifices of various sizes were placed at the entrance and exit of the volumes. The venturi and the orifice meters were used in these tests. It was found that it was possible with a volume of several cubic feet, combined with a pressure drop of about two inches of mercury, to reduce the error to a small figure, even for a meter having a large maximum error.

The Muffler as a Quieting Device. Following the experiments with the volumes and orifices combined as a means of eliminating the pulsation, the idea was further developed by the combination of a volume with several orifices; or in other words, the adaptation of the principle of the automobile muffler to the problem of pulsating flow. It was found that the effectiveness of any single type of muffler, aside from its value as a volume alone, depends entirely upon the amount of throttling produced and very little upon the design and arrangement of its baffle work.

Conclusions

The conclusions reached as a result of the investigation may be summarized as follows:

A Nature of Pulsations:

a Pulsations in a pipe line, originating from a reciprocating system, or a similarly disturbing system, consist of sudden changes both in the velocity and in the pressure of the fluid.

b The pressure change is the most apparent and is probably the greatest factor in producing errors in metering devices.

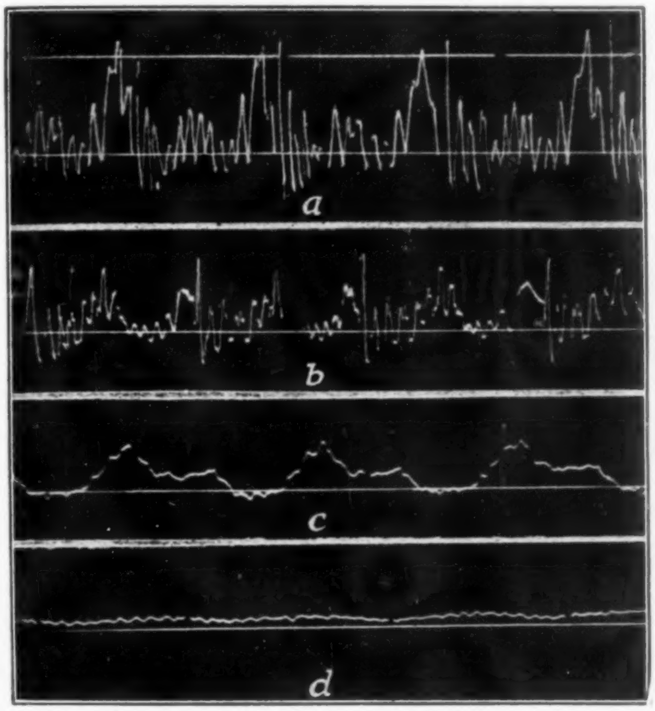


Fig. 6 Showing Quieting Effect of Pulsating Bag

(*a*) 54 ft. below compressor; maximum pulsation in line.
(*b*) 54 ft. below compressor; pulsating bag attached to quiet pulsations.
(*c*) 88 ft. below compressor; shows slight quieting effect of the 12-in. volume.
(*d*) 117 ft. below compressor; shows complete quieting effect of the 24-in. volume.

c The pressure change is in the form of a wave front resembling a traveling sound wave of low frequency.

d The pressure wave travels in the pipe with the velocity of sound.

e The velocity of the pulsation is independent of the velocity, or quantity, of fluid flowing.

f Pulsations in air flow are similar to the compression waves set up by water hammer. Both travel at the velocity of sound in the fluid and are independent of the velocity of flow.

g The effect of this pulsation on a flow meter is to increase its reading, often causing an error of great magnitude. The magnitude of this error depends upon the frequency of pulsation, nominal static pressure of the fluid, type of meter used and adjacent fixtures in the pipe line.

h With orifice meters and flange nozzle meters the pulsating error increases as the diameter of the orifice, or nozzle, approaches the diameter of the pipe.

i The throttling or modification of the manometer connections to the meter does not appreciably reduce the error.

j The point of attachment of manometer connection has no great effect on the error due to pulsating flow.

k The pulsation error at the center of the pipe is 35 per cent less than that at the wall of the pipe.

l A meter on a "dead-end" connection will usually show a positive error of considerable magnitude.

m The pulsation must be eliminated or greatly reduced in order to have the meter read without objectionable error.

B Practical Elimination of Pulsations:

n Because of the high velocity of the pulsation, an excessive length of pipe line would be necessary to destroy the pulsation.

o Throttling is effective but requires a pressure drop of 6 in. of mercury to reduce the error to 5 per cent.

p Abrupt volume enlargements in the pipe line will eliminate the error, if of sufficient capacity. A volume capacity of 20 cu. ft. is required for an error within 2 per cent.

q Generally speaking, for the same capacity, a volume of relatively large diameter is more effective than one of small diameter.

r No relation was found between the compressor displacement and the capacity of the volume chambers.

s The combination of throttling with volumes forming the "muffler" device probably is the most effective device for the mechanical elimination of pulsations.

t The pulsating bag, or diaphragm, and the fan, or revolving baffles, are partially successful in eliminating the pulsations, but their installation is thought to offer serious practical objections.

u The effectiveness of any of these quieting devices seems to depend upon their ability to dissipate or change the energy of pulsation which is effected chiefly through a drop in pressure.

v The device which will destroy the pulsating energy with the least obstruction to the flow of the fluid is the most desirable.

w The effectiveness of the meter element itself in quieting the pulsation depends upon the degree of restoration of the pressure beyond the meter. The greater the percentage of restoration, the higher the percentage of error shown for any given type of meter.

C Adjustment of Error of Pulsation:

x It is probably not feasible to correct any meter by means of a correction factor owing to the disturbing effects which may arise from slight changes in the installation and running conditions.

y The experimental establishment of a pulsating correction factor and its relation as shown in the formula proposed in the complete paper is not considered feasible with our present experimental knowledge of the laws of pulsating flow.

z It seems probable that each installation where pulsating flow is present would present its own peculiar problem for which an individual study and consideration of the existing conditions would be necessary for a satisfactory solution.

The complete paper contains additional illustrations of apparatus, indicator and photopulsometer diagrams, tables giving the data obtained in the investigation, and a bibliography of the subject. It also discusses the possibility of adjustment of errors at considerable length.

Discussion[1]

H. N. Packard,[2] who opened the discussion, said that he did not agree that the pressure change was the greatest factor in producing errors in metering devices. For instance, imagine a cylinder and piston to be discharging through a short length of pipe, including a pitot tube, into a large volume such as a gasometer. In the meter section no appreciable static pressure increase could be measured, but a very appreciable variation in rate of flow must occur with the consequent error of meter reading. As practically all meter installations were fairly close to the pulsation-producing piston, he believed the trouble caused was mostly due to actual instantaneous flow variations through the metering device.

The statement was made that at least a 6-in. mercury pressure drop was required to reduce pulsations to a practical limit. Did the authors consider this a general statement or applicable only to their test conditions? It would appear to him to be a function of the density of the fluid, its velocity and the pulsation wave form (magnitude of pulsation) if made as a general statement.

He was still of the opinion that there was some relation between the piston displacement and volume of a quieting receiver. Taking the two absurd extremes of a volume equal to piston displacement and an infinite volume, in one case it was known that no effect would be produced and in the other that perfect quieting of pulsations would occur. He believed that the quantity of fluid discharged per stroke, the number of strokes per minute, and the volume and diameter between the source of pulsations and the meter determined the pulsation effect at the meter, at least with elastic media such as gases.

On dead-end error tests he believed that there was an actual displacement of gas back and forth in the meter, this flow effect being due to the elasticity of the gas which was alternately compressed and expanded in the dead-end volume.

J. M. Spitzglass[1] wrote that prior to the advance of Professor Judd's experimental work on pulsating flow there was an idea prevalent that the error in the measurement was due mainly to the magnifying effect of the differential column, reading the average height and the corresponding square root of this average instead of the average of the instantaneous square roots which were the equivalent of the varying flow.

With the development of the flow meter, his company had sought to eliminate this error by making the meter respond electrically to the instantaneous instead of the average height of the differential column. This provision was thought to eliminate the part of the error which the authors of the paper designated as the "effect of the type of manometer used." They soon discovered that there was a much larger error due to the "harmonic" effect of the pulsations in the flow. Still, in all their observations with reciprocating flow this error had seldom exceeded 25 per cent under any circumstances. Furthermore, this error could be easily eliminated by moderate restrictions in the form of additional orifice plates on either side of the differential medium.

The effect of pulsation, according to Mr. Spitzglass' understanding of the investigation, was shown to be rather in the nature of an additional term than a factor in the algebraic expression of the flow for a given meter.

R. J. S. Pigott[2] submitted a written discussion in which he said that the problem of pulsating flow was largely an acoustic one. All the data went to show that the variability of the conditions was due to the fact that the acoustic conditions in the pipe differed with every installation, and it was hard to see how pulsating flow could be stopped in every case until a study was made of the phenomena from an acoustic standpoint.

One of the earliest problems in the opinion of the members of the Fluid Meters Committee had been that of either providing correction for the effect of pulsating flow upon the indications yielded by the flow-meter mechanism, or to so reduce the pulsations as to render their effect insignificant. The net result of research had definitely confirmed the belief that any type of meter operating on a difference of head which was proportioned to the square of velocity would register high on pulsating flow. The other belief, that it was very difficult, if not impossible, to provide suitable correction factors for the readings of the meter, was also very largely confirmed. The problem, therefore, was mainly reduced to providing commercially practicable means for suppressing pulsations to a point where they would not have a marked effect in the registration of the meter.

The experimental work so far carried out had indicated that it was feasible to accomplish this end by means of a combination of throttling with enlargement of volume. The scope of the experimental work had not been large enough as yet to definitely establish the amount of throttling and the amount of volume enlargement required for any particular case, and it was probable that the variations in velocity and size of lines would render an exact solution for any specific case difficult.

The first report of the Fluid Meters Committee was to have been produced for the Annual Meeting but the amount of work to be done both in editing the report and preparing for printing was

[1] These extracts from the discussion deal more particularly with those portions of the paper appearing in the preceding abridgment.

[2] Mech. Engr., Cutler Hammer Mfg. Co., Milwaukee, Wis. Mem. A.S.M.E.

[1] Vice-Pres., Republic Flow Meter Co., Chicago, Ill. Mem. A.S.M.E.

[2] Mech. Engr., Stevens & Wood, New York, N. Y. Mem. A.S.M.E.

too much to permit publication at that time. This report would cover the matter of installation very fully, as well as the theory and accuracy of the devices employed. It was to be hoped that it would provide for the designers and users of flow meters a complete summary of information available on the subject. Hitherto there had been no single source from which this information could be obtained, and it had been scattered through three or four hundred different publications.

John L. Hodgson[1] wrote that in his opinion, the results obtained from the elaborate researches described in the paper might have been very much greater had a careful analysis been made beforehand of the ways in which pulsating flows caused errors in meters which were based upon the measurement of a differential pressure. By making such an analysis he had found it possible to reach wider and more general conclusions than the authors and at the expense of far less experimental work. Some of the most important of these conclusions could be summarized as follows:

A pulsating air flow might be considered to consist of:

a A "pressure variation" which was transmitted with the velocity of the fluid in the pipe, plus the velocity of the sound in the fluid proper to the particular size and roughness of pipe used, and the nearness to the source of pulsation of the point where the velocity was measured.

b A "velocity variation" during which the whole of the air in the pipe was accelerated or retarded. The fluid at a point distant from the source of pulsation did not, however, change its velocity until the impulse, transmitted with the velocity stated under *a*, reached it.

Both these pressure and velocity variations caused errors in the meter, but in quite different ways.

The error due to the pressure variation occurred when the pressure pipes leading to the meter had different coefficients of discharge for inflows and outflows, and when the capacity in the meter on the two sides of the water or mercury column were different. It was then possible to obtain an actual difference of pressure on the two sides of the meter by the pressure variation alone and when there was no velocity variation at all in the pipe.

The error due to the pressure variation might easily be brought down to a very small amount by using pressure connections which had equal coefficients of discharge on both directions, and by keeping the capacities in the meter about equal.

There remained the error due to velocity variation, which was the real source of trouble. It could be shown by calculation that for certain wave forms this "velocity variation" might produce errors of several hundred per cent.

The error due to this cause could be calculated or determined by calibration for any particular conditions; but as it varied with the rate of flow, and the wave form, and the product of the specific volume and the absolute pressure of the fluid, and the loss of pressure in, and the capacity of the pipe line, it was best reduced to a small amount rather than allowed for.

The only way to reduce it was to smooth out the wave form of the "velocity variation" at the metering point. This could be done in many ways, the simplest of which (not mentioned by the authors) was to insert a capacity *and* a throttling device between the source of pulsation and the metering point. If the meter itself offered sufficient resistance it might form the throttling device; if it did not, an additional throttling device might be added. The capacity should be placed between the source of pulsation and the meter, and the additional throttling device, if any, should be placed on the downstream side of the meter.

In their closure the authors, replying to Mr. Packard, wrote that they could readily see that his type of meter would not be greatly affected, if any, by the pressure changes, even though the static pressure gage might read higher due to the pulsation. They would also conclude from their investigation that his meter would be less affected by the pulsation because the effect on the velocity head seemed to show much less error than that produced in meters depending on the pressure drop readings.

In regard to the effect produced by the static pulsation, they had failed to convey the proper meaning. The change or effect on static pressure produced by the pulsation was much greater than the effect produced on the velocity head. This pulsation, like sound, seemed to be propagated as a pressure wave and the effect produced on any measuring device, especially where difference in pressure head was used, was much greater than the effect recorded on the velocity diagrams from the photopulsometer. Hence, the conclusion that the "pressure change" was the greatest disturbing factor was drawn. This they believed to be born out in their work.

It seemed apparent that the pulsation (assuming its propagation as a pressure wave), was transmitted in the pipe by means of the air (either flowing or quiet) as a medium; and that with the dead-end meter connection the pulsation surged back and forth independent of the air which itself might also have some slight movement back and forth. This transmission of pulsation in the dead-end line would seem to be similar in this respect to the surge of pressure in a water line due to water hammer, which was very much greater in magnitude as compared with the effect due to velocity.

The conclusions given in the summary were made with reference to the installation which the authors had tested; also the reference to the throttling effects of a 6-in. mercury pressure drop considered their test conditions only, and should be modified much according to Mr. Packard's suggestion.

Referring to the relation to piston displacement of the volume of a quieting cylinder, it was probably true that *some* relation existed, but it had seemed to them that it would take such an extended investigation to establish anything approaching a law, as to render the solution impracticable.

In the dead-end meter installation, they agreed with Mr. Packard that there was an actual forward and back flow of the fluid due to the elasticity of the gas; but it was also true, they thought, that the pulsation in the form of the compression wave traveled forward in undiminished amplitude and returned in more or less diminished amplitude, depending on the length, shape, and volume of the dead-end connection.

Mr. Spitzglass stated in his discussion that the authors in finding the percentage of error due to pulsating flow had compared "a variable quantity, the pressure pulsations, on the basis of another and more variable quantity, the velocity pressure of the meter."

The error due to pulsating flow was based on the velocity, or quantity of flow, or its proportional equivalent the square root of the pressure difference through the meter element for pulsationless flow. For the four types of meters used the velocity head was equal to or proportional to the drop, or pressure difference, through the meter element. From whatever cause the pulsating flow might have been produced it was quite evident that its effect would have to be determined from the reading on the meter manometer.

It appeared to the authors, therefore, that while the pressure pulsation seemed to be the greatest factor in the error due to pulsating flow it was the velocity head reading that was observed on the meter. In their opinion it was the velocity-head readings as shown by the meter for both conditions of flow that should be compared. In fact, they were at a loss to know of any other way of establishing the percentage of error.

As pointed out in the paper and as further emphasized by Mr. Pigott, the authors believed that very little could be done to establish suitable correction factors for meters operating under pulsating flow and that the solution of the problem was reached when some suitable means were provided which would reduce the pulsations to a negligible point. The adaptation of the "muffler" device was apparently the most effective mechanical device for reducing the pulsations. However, further study and experimentation were necessary to establish the proper combination of throttling and volume space for static pressures and pulsating conditions approaching those in general practice.

Mr. Hodgson took exception to certain conclusions in the paper, in some cases justly so, and in others due apparently to a wrong interpretation. He pointed out the importance of having equal spaces in the manometer connections of the meter and in the case of the photopulsometer equal spaces above and below the diaphragm. The authors also recognized the importance of this and so far as possible all manometer connections were made of equal length, although this relation could not be maintained while the manometers were in use.

It was conceivable that the pressure pulsations might be lessened,

(*Continued on page 270*)

[1] Egginton House, Bedfordshire, England.

Some Engineering Aspects of the Design of Musical Instruments

By WILLIAM BRAID WHITE,[1] CHICAGO, ILL.

The object of this paper is to propound and answer two principal questions, namely, to what extent may the problems of producing musical tone by means of musical instruments be considered as engineering problems, and how far engineering principles and practices can be expected to work improvements in the efficiency of the most important musical instruments and in the economy and exactness of their manufacture. The major portion of the paper is devoted to the consideration of the specific problems presented in the manufacture of pianofortes.

ENGINEERING principles and practices are not at present inevitable concomitants of the making of musical instruments. Nevertheless it can be shown that musical-instrument manufacturing involves in all branches problems familiar to the mechanical engineer; and that if the principles of mechanical engineering were more generally adopted as the foundation of such manufacture, vast improvements would in due course be achieved.

Most of the present paper is devoted to a consideration of the specific problems presented by the manufacture of pianofortes, with some passing observations on player-pianos and organs. The first-named instruments occupy a position not only the most prominent in the music industries, but also the most favorable for the introduction and application of scientific ideas. Along with them, it is true, ought to be placed the various wind instruments of brass and wood, but these as yet are more important artistically than industrially; while the bowed instruments of the violin family are not practical subjects for scientific investigation from the engineering standpoint, or at least will not be such until their manufacture has reached the level of large-quantity production. Nor need there be, for the purposes of this paper, any consideration of the comparatively unimportant instruments of percussion and other minor contributors to the modern orchestra.

It is evident that any musical instrument whatsoever depends upon the recognition of, and obedience to, those laws of physics known as the laws of acoustics. Therefore it can easily be understood that scientific methods, based upon exact calculation, should form the foundation of musical-instrument building. In a word, musical-instrument building is one of the mechanical arts.

The modern pianoforte factory compares favorably with all other woodworking plants in respect to the adoption of machinery for cutting and finishing lumber and for fabricating it into wooden structures. But in regard to the tonal elements in pianoforte making, scientific reformation is imperatively needed and the present discussion is therefore confined to this phase of the problem.

The tone-producing elements in the pianoforte consist, first, of steel wires which are struck by "hammers" made of wood covered with felt. These hammers are swung on centers to which their shanks are pinned, and they move against the strings under the impulse of the musician's fingers conveyed through an elaborate system of levers, known technically as "the action." There are eighty-eight "unisons" or "notes," each consisting of two or three wires tuned to the same pitch, and each with its own hammer, its own system of levers, and its own finger lever or "key." With the rapid development of pianoforte playing since the early part of the nineteenth century, the indifferently drawn thin wires previously used had soon to be superseded by heavier material. The introduction of cast-steel wire, however, caused additional strains, which necessitated solider and stronger framework. Thus arose the now universal practice of carrying the strings of the pianoforte upon a frame or plate of cast iron. Herein lies the central engineering question of pianoforte making.

Another important tone-producing element of the pianoforte is the "soundboard." This is a sheet of spruce wood, built up from selected strips of the lumber into a square or wing-shaped table, according as it is to be used for a vertical (upright) or horizontal (grand) instrument. The strings pass over this resonance table and are put in contact with it by means of wooden bridges on which they rest, and which determine their vibrating lengths. Considering that spruce is a soft and not very durable wood and that the sheet must not generally exceed $^3/_8$ in. in thickness, it is evident that a rather heavy duty is placed upon a very slight structure in the carrying of the downward pressure of the stretched wires.

In practice the stretched wire is looped at one end over a hitch pin and then crosses the wooden bridge which connects it with the wooden soundboard. It then is stepped off at another bridge called the bearing bridge, from which it passes around the tuning pin.

Basic Conditions of the Problem

Now let the following facts be noted: First, the number of these strings or wires is about 230 in the standard pianoforte. The ten lowest (counting from the bass upward in pitch), which are also the longest, run one to a unison. The next twenty-five or so (differences exist among individual makes) run two, and the remainder, three to the unison. The tension exerted by all these wires when tuned to the standard pitch is usually not less than 35,000 or more than 50,000 lb., variations being due mainly to the individual ideas of various manufacturers as to the advantages of this or that degree of tension.

Second, the highest of these unisons (in pitch) utilizes strings each about two inches in length. At distances of an octave the string lengths are multiplied in the ratio 1:1.875, so that the increase in length from unison to unison (12 to the octave) is approximately in the ratio 1:1.054. Thus, in the largest instruments, the so-called concert grands of about nine feet overall length, the lowest bass string may be 8 ft. long. These length ratios represent the best contemporary practice, but are not necessarily binding.

Now a length of 8 ft. for the longest string on a pianoforte of this size does not, of course, represent a correct proportion, since there are seven octaves of tones. In fact, at certain points in the design of each style of pianoforte it is necessary to check the progressive lengthening of the strings, which would otherwise begin to exceed the length of the instrument, and instead to lay out each succeeding unison on the plan of increasing the weight of the wire so as to make up for the inability to increase the length as much as would otherwise be necessary. These overweighted strings are usually known as the "bass strings" or the covered strings, and form a distinct section of the design, though in a tonal sense closely related to the remaining unisons.

Third, the limitations of the human hand prescribe a total width of 48 in. for the keyboard, which is equivalent to the width of 52 white keys. The black keys are interpolated at intervals between and behind the finger plates of the white ones. Hence the entire layout of the strings, with all the arrangements for their support, must be contained in a structure which, in the largest grand pianoforte, may be as much as 60 in. wide at the keyboard end, and as much as 9 ft. long. In a small vertical (upright) instrument, the extreme width may not be more than 53 in. and the height 48 in. An ordinary grand of small domestic size will be perhaps 54 in. wide and 63 in. long.

The chief engineering problem in pianoforte design is to produce a supporting structure which will withstand the tension of the strings and provide complete stability under any normal condition of temperature or atmosphere, so that the strings will not slacken or tighten into an out-of-tune condition, and so that the soundboard shall be free to vibrate and to perform its duties as a general amplifier of the sounds set up by the vibrating strings.

Fig. 1 shows the string plan, with the bass or covered strings crossed over and above the others in order to secure the greatest

[1] Technical Editor, *The Music Trade Review* (N. Y.); Associate Editor, *The Talking Machine World* (N. Y.).

Abridgment of paper contributed by the Forest Products Division for presentation at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. All papers are subject to revision.

possible length. This is the sort of design which would be adopted for a small horizontal piano of apartment or domestic size, and is typical of present-day practice.

Tension Figures Calculated

The scale is divided into four divisions, and the strings in each have an approximate total tension as shown in Fig. 1. Assuming that the scale has been designed so that the average tension per string is 150 lb. in the three treble sections and 160 lb. in the bass section, and that there are three strings per unison to the three treble sections, then the tensions will be (proceeding from right to left): Upper treble division, 17 unisons, 7,650 lb.; middle treble division, 18 unisons, 8,100 lb.; lower treble division, 25 unisons, 11,250 lb.; which gives for the treble divisions a total tension of 27,000 lb. while for the bass section of 28 unisons of covered strings at 160 lb. per string, 18 of which are two to the unison and 10 one to the unison, we have 7,360 lb., making a complete total for the scale of 34,360 lb.

In regard to the problem of the supporting structure, the idea of a solid iron casting across which the strings shall be stretched and which shall take up their pull without putting any undue strain on the wooden soundboard, has been generally adopted since the middle of the nineteenth century. In general, the iron frames, or "plates," as the supporting structures are commonly called, are often far too heavy and the iron is frequently not well distributed. When they crack, as they sometimes do after the strings are first pulled up into tune, the first expedient is usually to thicken the broken member. A great deal of time, labor, and expense would of course be saved in these cases if the plate had been from the first designed by a competent engineer who could have calculated the stresses and strains and designed the tension, shear, and compression members accordingly. The relation of the strings to such a structure is incidentally shown in Fig. 1.

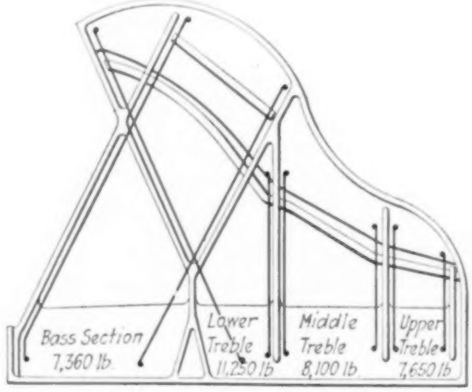


Fig. 1 String Plan and Iron Frame of a Grand Piano

These iron frames, cast in one piece, fulfil two main objects: they support the tension of the strings and limit the tone-emitting lengths of these strings at the ends nearest to the tuning pins, while affording hitching places for them at the other ends. They thus consist in principle of three parts, namely, a tuning-pin plate, through which the tuning pins are driven and which is under a shear strain; a hitch-pin plate at the other end which contains the hitch pins on which the ends of the strings are fastened; and a series of compression members or struts which serve to keep the two plates at the proper distance from each other when the strings are under tension, at the same time preventing the structure from buckling. Fig. 1 shows the hitch-pin plate at the top, and the tuning-pin plate at the bottom.

Description of Supporting Structure

Fig. 2 shows the outline of a supporting structure designed to take up the strains described in the case discussed in connection with Fig. 1. It will be seen that for the bass section two struts are used, one of which has been numbered 2 and 3 and the other one 5. In the tenor or lower treble section there is one long diagonal strut marked 1 and 4 and one almost perpendicular marked 6. Struts 7 and 8 delimit the extreme treble section.

Now in most cases of current practice the struts are cast in rectangular section and are always deeper than they are wide. The cross-section of No. 3 is usually about 1.75 sq. in. in area. The cross-sectional area of the lower or hitch-pin end of the same strut is always less, owing mainly to the need for crossing it over the wooden string bridge with necessary clearance; 1.25 sq. in. is the area usually employed for the cross-section for No. 2.

The limiting strut at the lower end of the lower treble is usually designed at about 1.75 sq. in. for the part designated No. 1 and 1 sq. in. for No. 4.

Strut No. 5, limiting the upper end of the bass section, is often made with a cross-sectional area of 1.25 sq. in. and Nos. 6, 7, and 8 might be put down at an average of 1.25 sq. in. each.

It is evident that if a modified T- or I-bar construction were used, the cross-sections could be considerably smaller and the plate a good deal lighter. It is also plain that there is a torsional movement and a bending movement caused by the upward pull of the string as it crosses over the bearing bar near the tuning pin and is pressed downward under the agraffe or pressure bar, by means of which it is limited at the tuning-pin end and held tight against slippage when in tune. Fig. 3 shows the nature of this bearing.

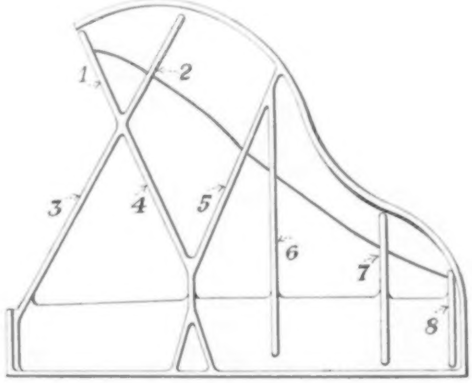


Fig. 2 Strut System in the Iron Frame of a Grand Piano

This torsional strain, and the upward pull as well, make it necessary of course to stiffen the struts rather more than would be necessary if they were to be used solely as compression members.

It should now be clear that the problem of designing the supporting structure of the pianoforte does merit the attention of engineers. In one sense of the term, the frame or plate may be deemed analogous to a truss bridge, with the differences that the load does not change rapidly and that the vibrations set up are relatively insignificant as regards any effect upon the stability of the structure.

The great need is economy of material, of resisting power, and of design, to the end that the weight of metal may be reduced, the appearance improved, the standing-in-tune qualities maintained, and duplications of the original pattern be rendered accurate and facile. It is probable also that better engineering practice would produce marked economies in the cost of production and vastly improve the general tonal standard, by securing greater uniformity and accuracy in the foundational structure of the instrument.

Acoustical Side of Tone Production

The design of the iron plate leads to a question of quite equal importance, namely, the acoustical design of the tone-producing elements or strings. Musical sounds are distinguished one from another by their pitch, relative intensity (loudness), and color or quality. The latter property, which distinguishes the sounds of one instrument from the sounds of the same pitch produced by another, exists for each family of instruments in varying degrees. Pianofortes differ among themselves greatly in quality. Some emit tones which may be described as "rounder," "fuller," or more "mellow," or again "brighter," than those of others. The differ-

ences depend upon a variety of conditions, among the most important of which are:

1 Thickness of the wire
2 Density and molecular structure of the wire
3 Tension at which the wire is stretched
4 Nature of the material with which the wire is struck
5 Point of the string at which contact takes place
6 Velocity of the hammer which strikes the blow, which is of course a function of the finger impulse upon the key.

Basis of Uniform Tone Quality

To secure tone quality as nearly as possible uniform from end to end the designer of a pianoforte should secure substantially uniform tension in each string from end to end of the scale, and an accurately graduated progression in length and weight of strings from the extreme treble to the lowest bass. The practical obstacles in the way of the attainment of this ideal are: (1) the limited number of available thicknesses of wire, (2) the limits placed upon the available lengths of pianofortes, (3) the limits beyond which bass strings cannot be loaded without destroying their tonal efficiency, (4) the fact that music wire vibrates most readily and efficiently for musical purposes at a tension equal to about one-half of its breaking strain, and (5) the uneveness of the felt used in making hammers.

To find the tension T in pounds at which a given string must be stretched so as to emit a sound of given pitch in vibrations per second the formula $T=P^2L^2M/675{,}000$ may be used, in which T is the tension required, L the length in inches, M the weight in grains (avoirdupois) per inch, and P the pitch in vibrations per second.

The design of the bass section is rather more complicated than that of the treble because the lengths cannot be determined at pleasure, but are strictly limited by the size of the pianoforte. In fact, the only rule is to make the bass strings as long as the instrument will allow, and then to calculate the weight each should have in order to give the required tension. It is better, practically speaking, to make the bass tensions a little higher, say 160 lb. per string. With this knowledge, together with the pitch in vibrations per second of each unison and the predetermined string length, the weight can be calculated from tables which have been worked out by the American Steel & Wire Co. From these same tables the nearest combination of core (steel) and covering (steel or copper) to give that weight of wire may be read off.

This is the general acoustical problem in regard to the strings themselves. The design of the supporting structure, however, can never be scientific until the string plan is just as scientific. Uniform tension properly graduated length, and carefully calculated weight are thus essential to the higher possibilities of pianoforte manufacture.

The Wooden Framework

In connection with the foundational structure there is one element which has not yet been touched upon, and which, although it is a matter of woodworking, nevertheless demands engineering attention. The strings are stretched across a soundboard and supported by the iron plate; but this plate and this soundboard must themselves be supported at their margins. The necessary support is given by a framework of wood to which, in the upright, the tuning-pin block, the soundboard, and the iron plate are fastened, and to which also are attached the sides and key bed of the instrument. In the horizontal instrument the curved or wing-shaped case, built up of veneers and bent into shape in a continuous rim around a caul, is braced by a system of beams, on which are laid the soundboard and the iron plate. Questions of stress and strain calculation naturally enter into the design of these.

Engineering Qualities of the Touch Mechanism

The practical engineer, looking at the constructional features of the pianoforte, will discover two other elements affording opportunities for improvement, namely, the touch mechanism and the stringing system. The train of delicate levers which conveys to the hammer the slightest or the boldest movement of the finger upon the balanced key lever consists of small pieces of wood pivoted on german-silver pins centered in bearings bushed with a very thick woven special cloth. Contact points are faced with buckskin or felt. There is no lubrication, and yet the total resistance to be overcome by the finger never exceeds 2.5 oz. and is usually less than this. Probably the action is the last part of the pianoforte which need engage the attention of the engineer, since it is the one part to which engineering principles have already been successfully applied.

Parallel observations may be made concerning the system of stringing. This relates to the methods commonly used to fasten the tuning pins so as to enable them to bear the torsion of the strings wrapped around them, while allowing the tension to be increased or decreased at will for tuning. The common method is to use a slim steel pin about 0.3125 in. in diameter, almost uniform in thickness from end to end, furnished with an extremely fine thread and turned into a hole drilled in a plank or block made up of cross-banded maple veneers. The total length of the pin does not exceed 2.5 in. of which about 1.75 in. is driven into the block. Many methods have been suggested and tried, successfully from the mechanical point of view, of which the object has always been to substitute a mechanical system for the crude wood block and frictionally held pin. But trade prejudice has always prevented their adoption.

Present Position of Musical Pneumatics

The rise during the last twenty years of mechanisms for playing the piano either entirely automatically or under personal control through the agency of perforated tune sheets known as music rolls,

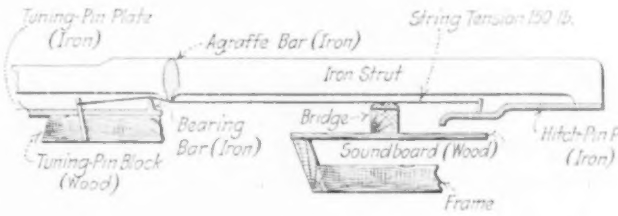


Fig. 3 Showing Method of Holding Strings Tight Against Slippage

introduces a new and fascinating set of engineering questions. These mechanisms employ, in all their forms, the pressure of atmospheric air against subnormal air pressures induced in closed chambers. That is to say, they are vacuum machines. The engineering problems they practically present may be grouped under three headings, namely, the production of the vacuum power, the prevention of leakage, and the design of the moving parts. In all these features many improvements are greatly needed. The present piano-player mechanisms are relatively crudely put together, depending upon glued-up plywood panels, rubber tubes connected with metal nipples cemented into wooden boards with shellac, and other similar devices which are clumsy, bulky, and very hard to standardize. The production of an all-metal action has been attempted, and with some success, but in practice the industry prefers to use wood, rubber hose and rubberized cloth, leather, and glue.

Problems of the Organ

The other great and complex musical instrument is of course the organ. In its many shapes, whether designed for ecclesiastical purposes, for the theater or for the home, it represents a vast complication of pipes, chests, electric contacts and cables, electromagnets, blowing engines and stop mechanisms. Unfortunately, however, the opportunities for engineering improvement are confined to the following points:

1 Blowing engine
2 Construction of the chests on which the pipes are set, with special reference to the possible use of concrete and metal in place of wood in order to eliminate the influence of temperature and atmosphere
3 Improvement in the electric mechanism which opens the pallets under the pipes when keys are depressed. Such improvement is mainly needed in order to promote durability and reliability

4 Interchangeability and standardization of parts.

These questions of interchangeability and standardization of methods, parts, and designs are the burning questions in musical-instrument manufacture. The great instruments which have been discussed are highly organized and comprehend an immense variety of parts and processes. The pianoforte draws its materials from mine, foundry, forest, sawmill, varnish factory, the sheep's back, the wire-drawing mill, from the elephant's tusks and from the sap of the caoutchouc tree.

The annual output of pianofortes is probably not to be valued on the basis of wholesale prices at more than $60,000,000 in an ordinarily good year. The industry needs, and would respond to, energetic and practical treatment from the standpoint of approved engineering practice.

Discussion at the Forest Products Session

At the Forest Products Session of the Annual Meeting of the Society, the following papers were presented: Some Engineering Aspects of the Design of Musical Instruments, by W. B. White; Lumber Dry Kilns, by Thomas D. Perry; Lumber Standardization, by F. F. Murray; New Factors Which Are Influencing Woodworking Machinery Design, by S. Madsen; and Control of Lumber Cutting Waste and Production, by C. M. Bigelow. The following abridged account comprises the discussion of the papers of this session that have already been published in Mechanical Engineering.

Discussion of Paper by Mr. White

Thomas D. Perry[1] opened the discussion of Mr. White's paper by asking the author if scientific investigations had been made of the resonance qualities of different woods used in the manufacture of musical instruments.

Mr. White replied that Dr. Miller, of the Case School of Applied Science, had made many valuable investigations during the past twenty years on the vibration of wood. He said that it would be highly desirable to know more about the behavior of a steel wire with reference to wood vibration and the effect on the vibration of wood of processes of preparing the wood for use.

Frederick F. Murray,[2] who said he was interested in the subject from the standpoint of the standardization of lumber, asked if wood as raw material was treated in the piano industries on the same basis that steel was treated in other industries.

Mr. White replied that the piano industry had grown from small beginnings 150 years ago until today it was including many machine and duplicate processes. The industry was still divided into a few branches, and in each of these the skilled workman did every operation belonging properly thereto. As long as such conditions existed, he said, it would not be possible to get standardization and interchangeability in the industry.

Mr. Murray said that in the furniture industry specialization had resulted in departments devoted to the manufacture of a single type of furniture, such as tables or chairs. In such shops the present standard classification of lumber was inadequate, and there was great waste resulting from piecework in shops operating on a production basis.

Mr. White spoke of the changes in the piano industry resulting from the demand for small grand pianos of the apartment size and for self-playing pianos. In all important shops, he said, there was a general standardization and a virtual interchangeability of parts. On the other hand, he said, the piano industry was essentially an art industry and the problem to be solved combined the ideals of art and production.

Mr. Murray said that standardization as he was interested in it did not mean a standardization of product nor an attempt to conform public taste to a standardized product, but dealt with the standardization of the raw material which would still admit of the most complicated fabrication. The purpose of such standardization was to eliminate the waste which now existed because of the lack of it.

[1] Grand Rapids Veneer Wks., Grand Rapids, Mich. Assoc-Mem. A.S.M.E.

[2] Hardwood Mfrs. Inst., Chicago, Ill. Jun. A.S.M.E.

Discussion of Paper by Thomas D. Perry

The discussion of Mr. Perry's paper on Lumber Dry Kilns[1] was opened by Kenneth Redman[2] who wrote that the paper inferred that the blower or forced-circulation type of kiln was suitable only for the drying of thin stock, or veneer. As a matter of fact, he wrote, the fan type of kiln was today meeting with its most marked success in the drying of extremely heavy stock which heretofor it had not been possible to dry in the pipe-coil kiln without the liability of enormous losses in the drying process. An examination of some of the fundamental facts to be considered in drying lumber would clearly show why this was possible.

Mr. Redman heartily agreed with Mr. Perry that the ideal way to dry lumber was to keep the surface pores open and to draw the moisture out of the wood at the rate that the moisture could transfuse from the center to the surface. In order to secure the maximum rate of such transfusion it was necessary to keep the outer pores of the wood open, and his experience confirmed the theory that the passing of comparatively large volumes of air so conditioned that each cubic foot would absorb only one or two grains of moisture, was by far the most speedy and economical. In other words, the fan type of kiln did not attempt to completely saturate the air, thus causing it to cool, become heavy and drop, and in this way create a circulation.

This feature of the pipe-coil kiln, regardless of its particular type, depended upon the varying density of the air in order to stimulate circulation, and consequently the drying power of moisture deficit varied directly with the length of time that the air remained in contact with the lumber. The air should be kept in contact with the lumber for as short a period as is commercially practicable so that each individual board in the pile of lumber might be subjected to air of practically the same drying power. The shorter the air travel, the more uniformly dry would be the lumber.

The principal duty of air in any drying process was to act as a conductor of heat which was transferred to the material to be dried, thus causing moisture to vaporize and then to be carried away. It was much more efficient to move air by means of a fan than to depend upon the heating and cooling effect of air.

A. A. Cutler[3] wrote in part that it would be unfair to a person wishing information on lumber kilns to let the author's statements concerning blower kilns go unchallenged. It was impossible, he wrote, to dry lumber at an even rate or in the least time unless the speed of air circulation in the kiln was rapid enough to bring each cubic foot of air in the piles back to the conditioning point before it had lost enough of its heat energy or increased its relative humidity to the point where its drying capacity was appreciably less than when it started on its circuit of drying, and nothing but a rapid circulation would accomplish this. The prevention of "interstrain" was simply a matter of conditioning the air entering the lumber piles.

If the circulation was sluggish and slow as in gravity circulation kilns, it would start to rise in the line of least resistance until the upper part of the compartment had been filled with air of the same or nearly the same temperature. Then it would gradually steep into the spaces between the lumber. As soon as it became cooler, by reason of having taken on some moisture, it would fall slowly until it got into a level of equal temperature or less humidity, where it would remain until by gradual change of kiln condition it reached the outlets at the bottom as in the case of Mr. Perry's recommended type of kiln. Hot air going up outside the pile and cold air coming down inside surely would not dry lumber evenly.

High temperatures and high relative humidities would reduce the moisture content of lumber by vaporization of moisture without creating any surface strains, but these two conditions could not be created evenly through piles of lumber without rapid circulation, and rapid circulation could not be created by anything less than a blower.

Mr. Perry had mentioned an actual operation schedule tested by practice of reducing one-inch oak from 35 to 5 per cent moisture content in 16 days. A properly constructed kiln of the blower type could do this same feat in eight days and this difference in drying

[1] Published in Mechanical Engineering, February, 1923, p. 110.

[2] Mgr., Dry Kiln Dept., B. F. Sturtevant Co., Boston, Mass.

[3] Cutler Desk Co., Buffalo, N. Y.

time was due to being able to maintain ideal drying conditions all the time in all parts of the piles.

The idea of steaming lumber in the initial drying stages, of course, did tend to even up any difference in moisture contents between the outside and inside of the lumber, but it had a very much greater value in heating the lumber to the kiln temperature. Except under conditions of 100 per cent relative humidity, the temperature of the lumber was always lower than the kiln temperature in proportion to the speed of drying. The employment of high humidities occasionally during the drying would speed up the drying time, but only in rapidly circulating kilns could this be done evenly through the piles. Within the ordinary range of kiln temperatures no damage could be done if the humidity was maintained at or near 100 per cent, but damage might be done when the humidity was reduced too rapidly. Again, it became obvious that control of humidity was very important, but how could it be accurately controlled if the circulation was so slow that widely different conditions must be created in one part of the pile of lumber before the air in another part would move out. In a blower kiln, with a speed of fourteen changes of air per minute between layers of lumber, the humidity and temperature would not have a greater difference than five per cent and five degrees, respectively.

Mr. Perry's assumption that blower kilns created too rapid surface drying was not in accordance with the facts. There might be some that did, but there were certainly others that did not, and association of rapid circulation with "too rapid surface drying" was unfair. It must be perfectly obvious that the drying rate was determined by temperature and relative humidity and that these two, so far as the conditions inside the piles were concerned, were controlled by the speed of movement of the conditioned air. Therefore, the rate of surface drying was under better control in a blower kiln than in a gravity circulating kiln.

Burritt A. Parks[1] wrote that he would like very much to have the discussion bring out more explicitly the relative advantages of the blower type and ventilated type of kilns. Among the advantages claimed for the former type were the following:

1 If the kiln is properly loaded, using end-piled trucks, a more uniform circulation of the air was obtained across both surfaces of the lumber, resulting in more rapid and uniform drying.

2 With practically all the air being recirculated, the temperature and humidity throughout the kiln were more uniform and more readily controlled.

3 Rapidity of circulation under absolute control, where a steam-engine-driven fan was used, making it possible to control kiln conditions in accordance with stock to be dried.

Some of the ventilated and condenser-type kiln advocates claimed great advantages for their kilns over the blower type on account of no power being required. This had always appealed to him as "sales talk" as the exhaust steam from a steam engine, used to drive the fan, could be used in the heating coils with a loss of only 10 to 15 per cent of its heat value in passing through the engine.

In answer to the discussers advocating the blower type of kiln, Mr. Perry said that it was obvious that there was no hope of his agreeing with them or of their agreeing with him, but he would like to call attention to two fundamentals which Mr. Redman and Mr. Cutler had not fully comprehended: First, that the maintenance of high humidity in rapidly moving air was exceedingly difficult and rarely accomplished, and second, that he knew of no means whereby a cubic foot of air would absorb only two or three grains of moisture and no more. It was his experience that the air would absorb more moisture than was desirable.

H. L. Henderson[2] asked the author what effect the periods of steaming and stewing had on the lumber. He also asked if the drying schedules put out by various kiln companies were based on moisture deficit in relation to the kind of wood being dried or if they had been worked out by experiment for a particular type of kiln.

Mr. Perry answered that very few data on the use of moisture-deficit curves for drying had been published. He believed this to be a wonderful field for the further development of scientific methods of drying. The steaming period was one which produced

[1] Byron E. Parks & Son, Grand Rapids, Mich. Mem. A.S.M.E.
[2] N. Y. State College of Forestry, Syracuse, N. Y.

a complete saturation of the lumber, a complete steaming at the temperature of the room. As the process was completed there occurred another which might be better termed a cooking period, which was a preparation for the more active removal of moisture during the drying period.

Anthony S. Hill[1] asked how it was possible to know that every piece of wood in every pile had been thoroughly steamed until its center was warmed.

Mr. Perry answered that the variable elements to be controlled in a dry kiln were humidity, circulation, and temperature. Nearly every dry kiln, he said, had means for the control of humidity by the admission of water or steam to the atmosphere of the kiln. Circulation was, of course, quite clearly related to temperature and would depend on the difference in temperature inside and outside the kiln if it were of the accelerated-draft type, so that circulation and temperature had to be controlled in relation to each other. Personally he was an ardent advocate of steaming lumber. He referred to the boiling of a potato as a means of removing water from it as being in a way analogous to this process in the drying of lumber.

A. A. Hemlen said that the author had pointed out that surface moisture could be removed at almost any reasonable speed.

If it was not known how fast water could be removed from a cell without destroying it, however, the surface of the drying problem had not been scratched, he affirmed.

Mr. Perry admitted that there was much about lumber drying which was not known. The practical rule-of-thumb measure was injury to the wood, and it was known that unless a uniformity of moisture content was preserved, injury would result.

T. Cassidy said that in his plant in which the very finest woods were dried, there were kilns of almost every type, and that with good operators and careful operation, good results could be obtained in any one of them. It was a question for engineers to settle, he said, which drying method would bring about the best results in the shortest time.

R. B. Wolf[2] who acted as chairman of the meeting, asked if there was a practical method of controlling humidity conditions in a kiln.

Mr. Hill said that humidity control instruments were available, but Mr. Murray pointed out that an automatic control by such instruments was difficult if not impossible.

In closing the discussion, Mr. Perry spoke of the need of research and development in the field of lumber drying.

Proposed Classification of Oil Engines

IN A paper on High-Speed Oil Engines, which was read by J. L. Chaloner before the Institution of Automobile Engineers in London, on February 15, and abstracted in *The Engineer* of February 23 (p. 202), the author said that while it has been suggested that there are too many types of oil engines to permit of a simple classification, nevertheless a detailed study of the many designs brings out the fact that they are in almost every case a modification of some fundamental principle or a combination of two or more such principles.

"Otto" has fallen into obscurity, "hot-bulb" requires qualification, "Diesel" is perplexing and inaccurate, "semi-Diesel" lacks precision, "cold-starting" is misleading, "high-compression" may refer to oil or gasoline engines, "crude oil" shows misrepresentation, and "refined oil" confesses a deficiency. Yet all the existing types can be allocated in accordance with definite ranges of pressure—whether compression pressure or maximum pressure is immaterial—and methods of fuel injection. When including high-speed engines the problems of nomenclature are certainly somewhat more complicated, because there is the tendency of the "constant-volume" and "constant-pressure" cycles to merge into each other. Mr. Chaloner accordingly suggests a system of classification of liquid-fuel internal-combustion engines under the headings (1) Gas Injection, (2) Air Injection, and (3) Mechanical Injection, each of which comprise (a) low-pressure, (b) medium-pressure, and (c) high-pressure engines.

[1] A. Hill Co., New York, N. Y. Assoc-Mem. A.S.M.E.
[2] R. B. Wolf Co., New York, N. Y. Mem. A.S.M.E.

The Airship for Long-Haul Heavy-Traffic Service

By RALPH H. UPSON,[1] DETROIT, MICH.

The factors upon which the value of the airship as a carrier depends form the subject of this paper. The airship is compared with other carriers, especially the steamship, and questions of speed, route, cost of transport, and time value are considered. Design and construction methods are discussed, as are also the problems of stability, dynamic lift, mooring, and fire risk. It is not the intention of the author to treat the subject in detail, but rather to indicate the possibilities of the airship for commercial traffic and to point out the chief problems toward which the engineer should direct his attention.

COMPARING airships with other common carriers, the most closely related is the steamship, of which the airship is a logical namesake. The fundamental difference is simply one of size and weight—the boat being "inflated" with air and other materials, the airship with hydrogen or other light gas. Take any boat which floats in the water, and enlarge its linear dimensions roughly ten times, *keeping the total weight the same;* it will then float in the air and may be called an airship. But just here is where the first big problem, that of structural weight, is met. For if in this example a similar design of structure throughout is assumed, it is clear that the airship hull must be made ten times as heavy to retain the same strength as its prototype, the boat, without even considering the l/r requirements of the compression members. At this rate the airship would not even "lift" itself, and this seemingly damning fact is perfectly true as far as it goes. But fortunately its effect may be modified or even reversed by other factors which may be utilized for cutting down the structural weight.

All the opportunities for lightening the hull structure must be taken advantage of to achieve a good weight-carrying efficiency. The first of these is the possibility of arranging the loading of an airship so that tension stresses predominate in its structure, whereas compression stresses must naturally predominate in a steamship. If properly utilized, this is a great factor in the final result. In a typical Zeppelin airship, for example, about 35 per cent of the hull structure is fabric, and about 12 per cent is high-tensile wire with an effective strength two to three times that of the most efficient compression girders. Recent improvements in the shape and arrangement of the hull permit a still higher proportion of tension elements.

Next, there is a big advantage in the fact that an airship is entirely immersed in the elastic fluid in which it floats, which insures a practically uniform buoyancy throughout. The effect is modified somewhat by vagrant currents, sudden gusts, and aerodynamic instability (which will be described later), but on the whole an airship in flight is subject to outside forces much less in magnitude and at the same time more positively determinable than those affecting a steamship in a heavy sea.

It is not only in engineering methods but in materials themselves that great advances have been and are still to be made. The first light-weight, heavy-duty gas engine was developed for airship use. In metallurgy the whole development of the remarkable alloys of aluminum for structural use owes its inception to the needs of airship design. A whole paper might well be devoted to duralumin, which seems destined to replace much of the steel now used in bridges, boats, railroad cars and all other structures where lightness is an appreciable factor.

But the greatest basic factor favoring airships still remains in the consideration of *resistance to propulsion.* Given a steamship and an airship of equal gross weight and equal speed, the latter will require only about one-tenth the horsepower of the former. Or, expressed in other terms, for an equal horsepower the airship will go over twice as fast; or assuming equal power efficiency for the two types, the steamship must be of a tonnage roughly 1000 times greater. This little-known fact is the airship's real reason for existence, for it is utilized not to save power but to gain speed. A speed of 60 m.p.h. for example, which is practically impossible for a displacement type of boat, is attained by the average airship with the greatest ease. Eighty-three miles per hour has been reached and 100 miles is simply a matter of design.

Until a very few years ago this great advantage existed only in theory. Before the war the greatest speed was in the neighborhood of 40 m.p.h., in a "ship" whose structural weight was about two-thirds of the total. The airship seemed a dead issue as far as any real commercial transportation was concerned. Now, however, we are taking almost full advantage of the theoretical resistance while avoiding almost entirely the disadvantage of extra weight.

In respect to size the airship is affected in much the same way as the steamship. The weights of various parts of an airship structure vary all the way up to the fourth power of the linear dimensions. Hence beyond a certain point there is an actual increase in the structural weight per unit displacement. Considerations of power and fuel consumption, however, bring the range of economical size far beyond that of greatest structural efficiency. This is because the

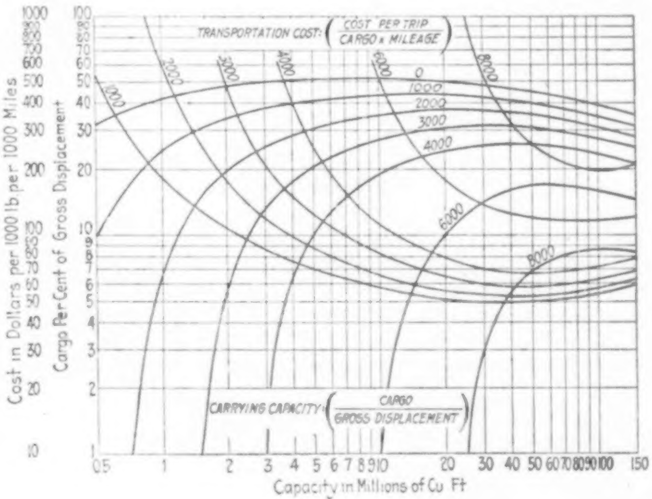


FIG. 1 EFFECT OF SIZE ON AIRSHIPS OF SIMILAR DESIGN, ASSUMING AN AVERAGE AIR SPEED OF 70 MILES PER HOUR

resistance, for equal speeds, varies as not quite the square of the linear dimensions.

All the above factors have been taken into account in computing the data for Fig. 1, which shows the effect of size alone on freight service over routes of different lengths. The curves that are concave downward show the net cargo weight as a percentage of the gross displacement weight, the figures on the curves themselves referring to the length of route in statute miles. The other set of curves (concave upward) shows the cost of transport in dollars per 1000 lb. per 1000 miles for the different routes marked. Note especially the range of sizes marked at the bottom in *millions of cubic feet.* Remember that the largest airship in existence today has a capacity of 2,500,000 cu. ft. and see how much further we have to go to use even to fair advantage this one item of size.

But this takes care of only three of the prime factors or variables entering into the consideration of a commercial airship line. Altogether there are five: namely, size, speed, route, cost of transport, and time value. The latter is the value of time, or of saving time, per unit of passengers or cargo carried. Taking the simpler case of freight (including mail and express), if time has any value it has a money value per hour on any unit weight of the cargo carried. This may be due to its intrinsic value, to perishability, news value, or other qualities. Let us take an assumed example. In a shipment of 1000 lb. of California fruit destined for the eastern states, an average of 20 per cent spoils during the trip. The remainder commands a price of 15 cents per pound. Suppose now that a saving of 40 hr. on the trip cuts the perishability in half and the fruit, being

[1] Chief engineer, Aircraft Development Corporation. Contributing editor, *Aviation.*

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fresher, sells for 5 cents a pound more. Here we have a total saving of $60 or a time value of $1.50 per 1000 lb. per hr.

Taking everything into account on a basis of present values, it seems probable that anything having a time value of $1 or more per 1000 lb. per hr. ($2 per ton-hour) can be shipped more economically by air than by rail, or about half of this figure in the case of shipping by water. This is assuming in each case the speed and size of unit best suited to the quality of goods carried.

Fig. 2 shows on a similar basis the comparison between all the available means of transportation. Here the speed is taken as net, with allowance for average stops, delays, time required to get in and out of terminals, and to load and unload. The shaded areas represent the respective economic fields to be served by steamship,

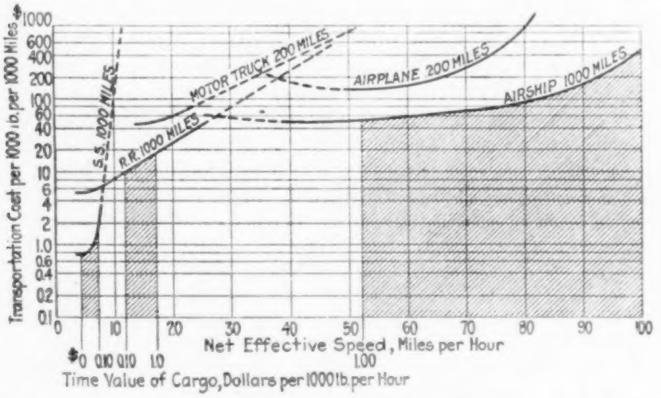

Fig. 2 Transportation Diagram Showing Economic Relations Between Different Carriers

railroad, and airship, which may be termed primary transportation units. The secondary units, motor truck and airplane, are not directly comparable with the others because their value lies in various special features peculiar to themselves. The airplane, for example, can combine high speed with small size and a comparatively short route, conditions which cannot be met by the airship.[1]

Design and Construction Methods

All lighter-than-air construction has developed from the free balloon as the basic type. Today there are five other more or less recognized types as follows:

- The kite balloon, which is the modern type of captive observation balloon
- The "blimp," a rather vague term usually applied to a small non-rigid airship with a single engine
- The non-rigid airship
- The semi-rigid airship
- The rigid airship.

The last three are only relative terms as all of them have much in common. For example, a non-rigid airship is usually stiffened as the nose in semi-rigid fashion. The usual "semi-rigid" ship is really rigid in principle, but its rigid structural parts are mainly concentrated in a keel at the bottom of the envelope instead of being partially distributed around it as in the Zeppelin "rigid." Again, it must not be thought that there is any appreciable difference in the rigidity of these different types in flight. The nomenclature merely has reference to the method of producing the necessary rigidity, the fundamental principles being the same for all.

Fig. 3 shows the Zeppelin type of construction, which is the only one so far used for the largest sizes. This consists of a sort of cage into which is put a series of large gas cells, the whole structure being covered by an outer fabric envelope. Here the non-rigid principle still exists but is confined to relatively small spaces between girders. The amount and distribution of structural elements is simply a matter of design based on the requirements to be fulfilled.

The design in general divides into two main parts: First, the aerodynamic design, in which the relations of speed, power, controllability and all external forces are worked out and correlated; second, the static design which takes care of all internal stresses and deformations, and distribution of lift and load. Calculations wherever possible are based on known laws and previous experience. But as an airship is so large and costly, and in so rapid a state of development, it is very important to have some way of checking calculations, for every new design, by experiments on models.

Problems Yet to be Solved

There are many problems connected with the control of airships which are as yet imperfectly solved. Quite contrary to what might be judged from general appearances, an airship in motion is naturally very unstable. It is not that it is subject to any sudden diving or loss of balance, for the mere mass of a large ship is enough to prevent that. Its instability is one of direction, i.e., it tends to keep on turning (either horizontally or vertically) in any direction in which it starts. Fig. 4 shows why. The air reaction or resistance at zero angle of course acts along the axis of the hull. But let the wind hit the model at even a small angle to its axis and a remarkable change occurs. The reaction is now much larger; its line of action is at a considerable angle to the wind stream, and crosses the axis far in advance of the hull itself, i.e., the hull is now acted on by a force which does not even touch it.

This instability may be made less serious by the mounting of fixed fin surfaces in the rear, but it is usually impracticable to attain complete stability in this way; so that we must still depend to a large extent on the rudders and elevators. The function of these is then not so much to turn as to keep from turning; in fact, a gradual turn requires the rudder to be held in just the opposite direction to prevent turning too sharply, and often to hold a given angle with the horizontal requires an elevator action opposite to what might be casually supposed. The development of satisfactory stabilizing means would take considerable strain off the pilot, make for a straighter flight path, minimize unpleasant motions, and facilitate mooring.

Another feature that may be appreciated by reference to Fig. 4

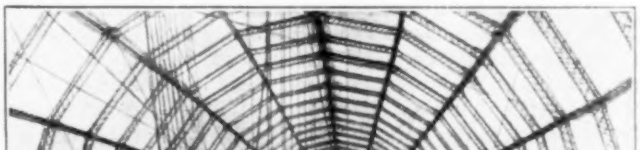

Fig. 3 Zeppelin Passenger Airship *Bodensee* Interior (Minus Gas Cells), Looking Forward

is the considerable proportion of dynamic lift that may be had at small angles of inclination. A 2,000,000-cu-ft. ship will lift in this way an *excess* load of 3 tons or more at full power. The lift is there—there is no doubt about that—but the problem is how to utilize it. An enormous airship cannot be started by running it along the ground like an airplane. It would be as easy to put wheels on an ocean steamship and run it on land. But some method of taking on the extra load and carrying it safely after the airship gets under way is at least a possibility and should be carefully investigated.

The way this dynamic lift functions now is generally in a reversed direction. The ship starts up approximately balanced, but changes in altitude and temperature are sure soon to cause variations in the lift which are compensated for by use of the elevators. These changes are not usually serious for a high-powered ship, but it is decidedly otherwise as regards the gradual consumption of fuel on a long trip. This is compensated at first by tilting the ship down

[1] Aerial Transportation of the Immediate Future, *S.A.E. Journal*, June, 1921, p. 593.

to get a negative dynamic lift, but eventually it becomes excessive even for straight flight and far more than is possible to land with. Then there are only two alternatives, either to let out gas or take on ballast. The latter, where possible, is the safer and cheaper. Two devices for this purpose are being developed at the present time; one is a trailing pump for taking water up from the ocean or other body of water; the other is an apparatus for condensing the moisture out of the engine exhausts. By the latter method more water can be collected than the original weight of the fuel, but the condensing apparatus so far developed is clumsy, heavy, and not entirely satisfactory. Another idea that has been used with some success is to burn the surplus hydrogen in the engines, thereby getting a reserve of fuel. This is a valuable recourse for emergencies, but hardly a substitute for the taking on of ballast, whose principal object is to prevent using up the gas reserve.

Another problem largely aerodynamic in character is the mooring and housing of a large airship. The resistance of a ship set approximately crosswise to the wind is commonly 30 to 40 times the amount of the head-on resistance. Hence the main principle of mooring in the open is to keep the nose headed constantly into the wind. In this position it will easily withstand any wind that blows.

The "mooring mast" as used in England, in which the wind itself is supposed to keep the ship in proper alignment, is not entirely satisfactory. In the first place, it takes great skill to bring an ordinary unstable ship up to a mast at all without breaking something. Then, even after it is moored fast, the ship will yaw through a considerable angle, putting corresponding strains on the structure. In a vertical plane the wind stays fairly near horizontal, but the ship does not, owing to the unavoidable changes in lift and trim. Furthermore, the ship is in a very inconvenient position for loading, unloading, inspection, and repair. But the most permanently serious problem of all those concerned with outside mooring may well be that of snow. A heavy snowfall without much wind would soon pile up a dangerous weight on the huge surface of the envelope and fins. Without other provisions for taking care of such a situation, the safest thing to do would be to cast loose and fly south with the birds!

Finally, any exposed mooring with the present rather perishable envelope fabric may be said to be an extravagence rather than an economy for any regular operation. Nevertheless, with metal envelope and other improvements the author believes that the time is not very far distant when an airship will remain in the open as a matter of course and only be brought into a hangar as a steamship is put into drydock, i.e., for general overhauling and special repairs.

Fire Risk

In addition to such problems, which range all the way from details to general construction, there is the fire risk, which seems to be the one considerable danger in airship operation.

It may be said at the start that the danger from fire is no doubt greatly exaggerated by many people mainly due to the accidents which befell the *R. 38* and the *Roma*. For airships there are no data by which we can estimate the risk on a percentage basis, for no regular passenger on an airship line has ever been lost, and no airship, aside from military and experimental ones, has ever burned

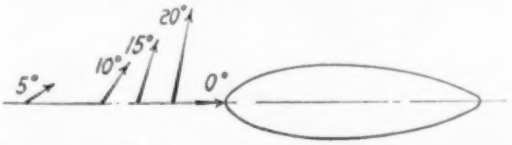

Fig. 4 Air Reactions on a Stream-Line Hull for Varying Angles of Pitch and Yaw

(Figures refer to inclination of wind stream relative to axis of hull.)

in the air. The Zeppelin passenger ships have carried approximately 40,000 passengers a distance of over 3,000,000 passenger-miles with never a fatality or serious injury. Nevertheless, the fire danger obviously exists and everything possible must be done to eliminate it.

In a present-day airship there are three principal inflammable elements: hydrogen, gasoline, and fabric. All three have possible substitutes, the most available being helium, heavy fuel oil, and metal, respectively.

The use of helium removes the largest bulk of inflammable material, but it is a mistake to suppose that the fire risk can be eliminated by this means alone. Hydrogen will not explode unless mixed with air, but it will burn if ignited by some outside agency at a point where it meets the air. For example, suppose a gasoline fire in one of the cars spreads to the envelope and ultimately reaches the gas. If the gas is hydrogen it will of course burn as it rushes out, at the same time rapidly accelerating the burning of the envelope. This is what happened to the front half of the *R. 38* after it broke in two. The use of helium would have been no insurance against the outer envelope burning away, and with it enough of the inner fabric to liberate all the gas; but the action would undoubtedly have been enough slower in that case to have saved many lives.

To eliminate the gasoline hazard, as well as to provide more economical fuel, there are several promising developments of heavy-oil engines now in process. This type of engine should be specially suited to airship work, which requires constant running for long periods of time.

To make the airship structure itself fire-resisting, the most feasible means seems to lie in a more extended use of duralumin. An all-metal envelope promises not only protection against fire but great improvement in durability and other desirable qualities as well. The metal ship of the future, especially if provided with heavy-oil engines, will be virtually fireproof as far as commercial service is concerned. It can thus be independent of the use of helium, which is rare, expensive, and deficient in lift. For military purposes, however, all three of the above-mentioned safeguards would be desirable.

Conclusion

To sum up the situation in general, it may be said that the improved airship will be definitely available for commercial traffic over either land or water in rough proportion to the extent to which the following conditions exist:

- Length of route (should be 500 miles or more)
- Density of traffic (at least 200 passengers or 50 tons of goods per trip)
- Time-value of pay load (passengers' time worth $6000 per year or more; goods $2 per ton-hour)
- Favorable meteorological conditions (no definite minimum).

The New York-Chicago route, which is so favorable in other respects, is the very worst in the country from a weather standpoint. A thorough analysis of this route by the Zeppelin Company shows the following results for present means and methods. A nightly 12-hr. service in both directions can be maintained 100 per cent on time for a season of six months. For the full year, however, the on-time trips would be cut to about 93 per cent. (An average of 4 per cent of the trips would probably be postponed or cancelled by advance notice, to enable prospective passengers to go by train if they so desired.)

A 60-hr. (average) transatlantic service could be maintained practically the year round.

The technical requirements of such a service are fairly clear, but the business side of it presents one of the hardest problems that have ever come up. In any other kind of transportation men have had the privilege of starting out with small units which, under favorable conditions, could at least pay their own way. With airships, however, it is necessary to choose between the frying pan and the fire. Either we must start with a comparatively small unit and a certainty of losing money on it or take a chance with large units on which we may lose a great deal more before they can be made profitable. It is indeed a great temptation to spend the larger amount of capital for what may look like good profits from the start. But it must be remembered that it is a new enterprise, entirely untried in this country; ships of the necessary size and construction have never yet been built; the whole system of safe and economical operation must be developed; and conjointly, the public must be educated to use the service. It is a maze of interreacting factors like the stresses in an airship hull itself, but much more indeterminate, and should certainly be as frankly recognized. A single ship of the smallest practical size (which is quite large and costly enough) to try out a given route will be well worth what it costs if it contributes but a small percentage to the "success insurance" of the larger enterprise.

Hydraulic-Transmission Variable-Speed Drive for Machine Tools and Manufacturing Processes

By WALTER FERRIS,[1] MILWAUKEE, WIS.

"Hydraulic transmission" has been in course of development as a variable-speed drive for twenty-five years, and has often been applied to gun control, turret control and steering gears on naval vessels during the past fifteen years. Its accuracy and flexibility of speed control, mechanical efficiency, durability of mechanism, and ability to stand abusive accelerating and stalling loads without overheating are well established, yet this type of drive has never attained extensive introduction in the industries, where its most extensive field seems to lie.

The author describes and illustrates a number of applications of the "Oilgear" to machine-tool driving, broaching, hydraulic presses, etc. This device is based on the same principles as earlier hydraulic transmissions, but has been developed from the beginning (in 1909) with a view to industrial requirements first, and automotive and marine applications afterward. The results obtained in practice during the past five years would apparently indicate that this method opens a field in which great improvements in machine-tool design may be made.

THIS paper is a report of progress made in applying "hydraulic transmission" variable-speed drive to machine tools and ordinary manufacturing processes, as distinguished from its previous applications to gun control and other naval work. In undertaking to develop a type of hydraulic transmission which should be commercially available, the designers of the device described later have purposely avoided the obvious field offered by the automobile, choosing instead the wide field of machine-shop drives and general manufacturing purposes. In this field the rigid requirements of extreme lightness and compactness do not apply, while the need of a satisfactory drive which will give any desired speed is just as great as in the automobile.

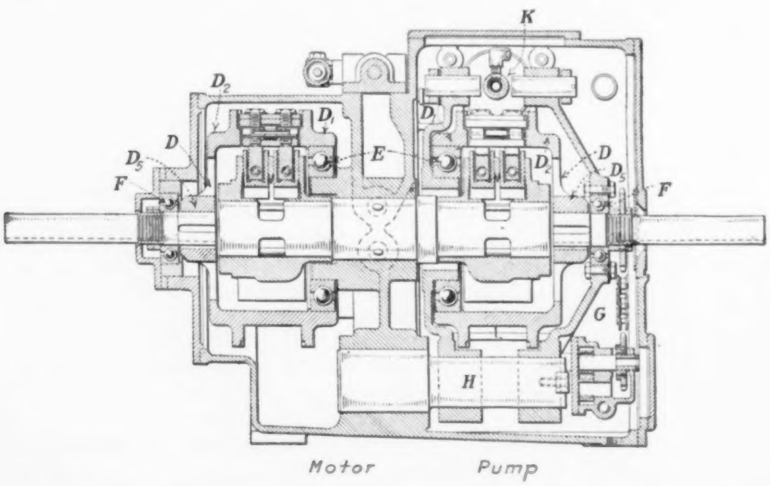

Fig. 1 Oilgear Hydraulic-Transmission Variable-Speed Drive, Longitudinal Section

Every hydraulic transmission comprises a pump, a motor, and a liquid circuit connecting them. For the driving of machines in machine shops and for general manufacturing we may add the conditions that the pump shall be a plunger pump, that it shall have a sufficient number of plungers to give a practically uniform flow, and that the stroke shall be adjustable at the will of the operator. Also, the only power-transmitting liquid herein considered is oil.

The various types of variable-speed hydraulic transmissions all employ multi-cylinder plunger pumps, and may be divided according to the arrangement of their plunger groups into axial and radial machines. Axial machines have cylinders arranged like the chamber of a revolver, parallel to and surrounding the drive shaft. Radial machines have cylinders arranged like the spokes of a wheel. In all cases there is an oil circuit having high-pressure ports and a conduit leading from pump to motor, and low-pressure ports and conduit from motor back to pump. In all cases several pump plungers are acting at once in communication with the high-pressure port, and several others in communication with the low-pressure port. This feature gives a uniform flow of oil. Some machines run with all working parts submerged in oil, and some with "empty case" and an oil sump below the level of the working parts. All have one member carrying a group of cylinders and another member operating the plungers in those cylinders, with the plunger stroke variable by the operator.

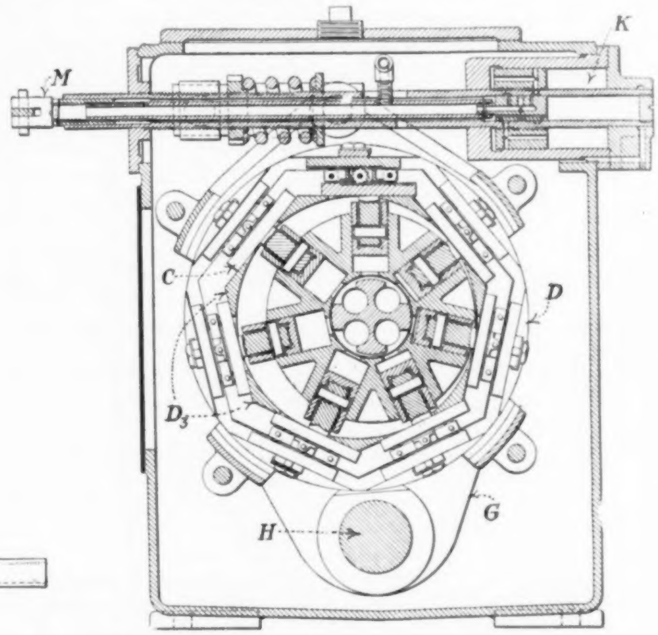

Fig. 2 Transverse Section Through Center of Pump

Design of the Oilgear

The Oilgear machine is of the radial type, its arrangement being shown in the longitudinal section (Fig. 1) of a unit containing both pump and motor mounted on a double-ended pintle and the transverse section (Fig. 2) through the center of the pump.

Fig. 3 shows the unit with two handhole covers removed, through which all of the plungers in both pump and motor can be withdrawn for examination if required. To remove a plunger it is only necessary to slide out a reaction plate A (Fig. 4) which is plainly visible through the open handhole, after which the roller-bearing cage B is removed and the crosshead with the attached plunger or plungers may then be withdrawn. It will be seen that each driver D (Figs. 1, 2, and 4) comprises two complete rings or flanges D_1 and D_2, integrally united by seven posts or bars D_3. Flange D_2 also carries hub D_5 into which the shaft is pressed, and the entire revolving driver D is supported on ball bearings E and F.

In the case of the pump these ball bearings are carried by a swinging cradle G (Figs. 1 and 2), which can be swung to the right or left around a stub or pivot shaft H. This serves to change the stroke along an arc of large radius, in place of the rectilinear slide used in the other radial machines. Figs. 4 gives details showing the plunger and crosshead unit with roller bearing, reaction plate, etc., in their relation to the driver. The thrust of the Oilgear plunger is

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delivered directly against the crosshead C, and thence through a roller bearing B to a corresponding reaction plate or roller path A carried by the driver D. There are no wristpins or rubbing bearings of any kind. The pressure on the plunger is limited only by the capacity of the roller bearing, which also reduces the friction almost to zero. The pins C_3 (Fig. 4) are not wristpins, but merely loose retaining pins to hold the parts together until the assembly

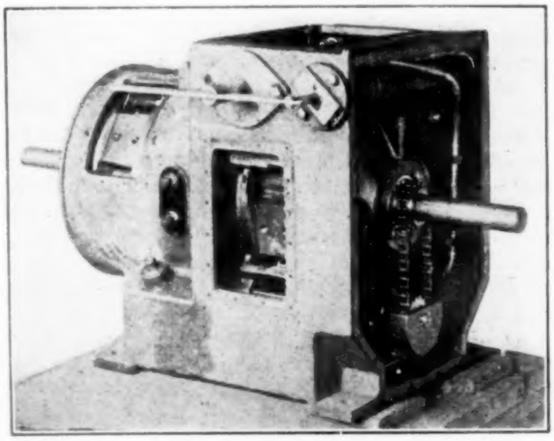

Fig. 3 Oilgear Machine with Handhole Covers Removed

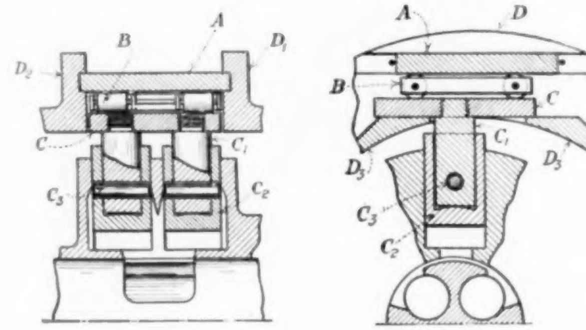

Fig. 4 Details of Plunger and Crosshead Unit

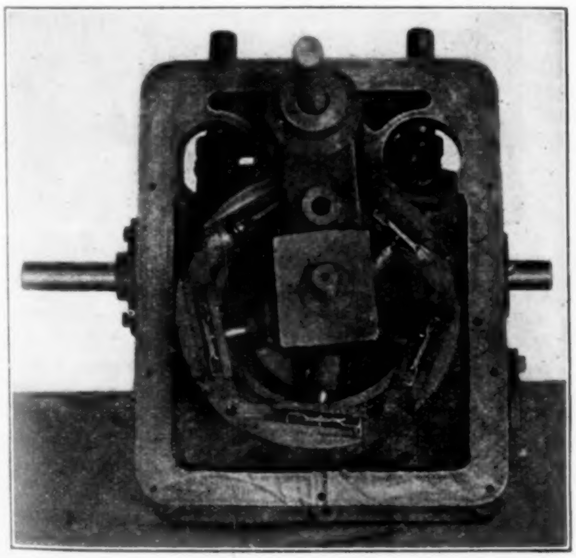

Fig. 6 Variable-Delivery Portion of QC Control Pump

is complete. All angular thrusts and side friction on plungers are eliminated. Figs. 1 and 2 also show the hydraulic stroke-changing mechanism K controlled by the pilot-valve stem M.

This paper is not intended to give a full discussion of the theory of the Oilgear, which in most points is similar to that of other hydraulic transmissions, but it is necessary to describe one special form before passing on to the applications. Fig. 5 shows a small Oilgear known as the QC control pump, connected to an ordinary double-acting cylinder operated thereby. Figs. 6 and 7 show the interior construction and working parts of the pump. The casing shown contains the pump only, the cylinder in Fig. 5 acting as the motor. This is in effect a hydraulic press, and it will be alluded to in that connection in a number of the applications subsequently described.

This control pump comprises the combination of a small variable-delivery pump, Fig. 6, with a large constant-delivery pump (the gear pump, shown at K in Fig. 7, and occupying the lower part of the unit) combined with it in the same casing and operated by the same drive and control. The machine also contains a distributing valve (J, Fig. 7) which is connected to the operator's handle on top of the case (Fig. 5). This handle may be moved from its central or zero position 90 deg. in either direction. During about

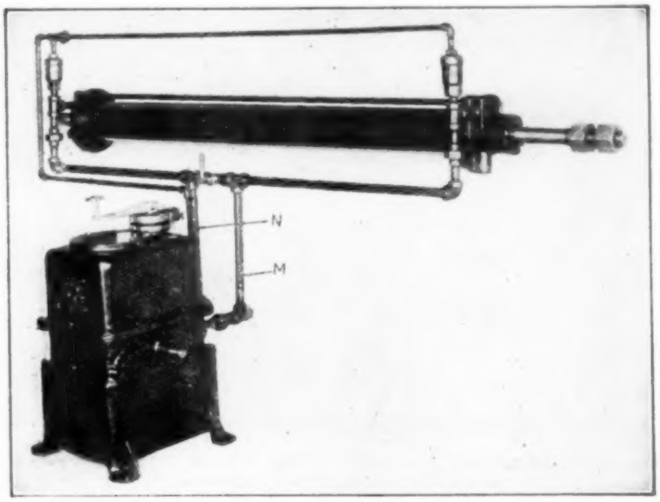

Fig. 5 QC Control Pump, in Casing, Connected to Double-Acting Cylinder

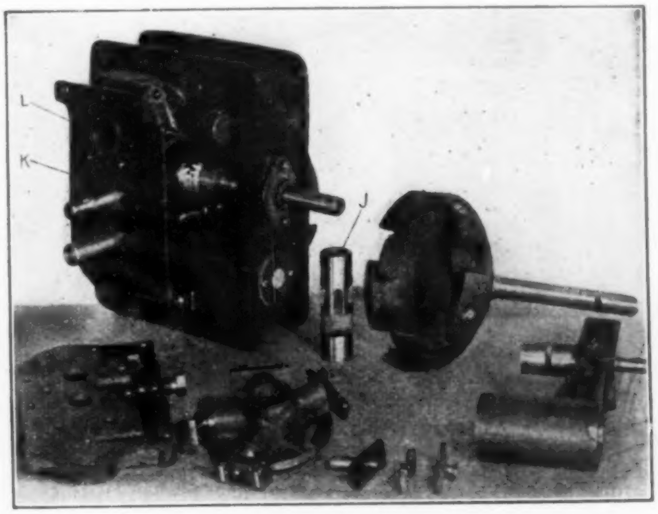

Fig. 7 Lower Part of QC Control Pump

60 deg. in either direction of this travel the effect is to change gradually the stroke of the variable-delivery pump, Fig. 6, thereby pumping more or less oil and moving the piston in the hydraulic cylinder shown in Fig. 5 either to the right or left very accurately, and faster or slower as the operator desires.

When the handle, Fig. 5, is turned 90 deg. either to the right or left to the extreme position, the distributing valve J, revolving in a ported sleeve L, Fig. 7, cuts out the variable-delivery pump from the main pipes M and N, Fig. 5, and connects the large gear pump K, Fig. 7, to these mains, thus driving the piston to the right or left with a very much faster movement. This serves to give a rapid traverse to a lathe carriage or boring-mill ram which may

be operated by this feed; or, in other cases, to the ram of a hydraulic press.

In a preceding paragraph the Oilgear hydraulic transmission was described as a combined unit in a case containing both pump and motor. Also the Oilgear control pump has been described, as made in a separate casing, for operating detached direct-acting pushing cylinders. It can also be used to operate detached rotary motors, such as the one shown disassembled in Fig. 8, which is adapted to drive the feed motions of machine tools, etc. Note the few and simple working parts, all made by ordinary machine-shop processes.

Machine-Tool Feed Applications

The principal elements are the QC control pump, shown in Figs.

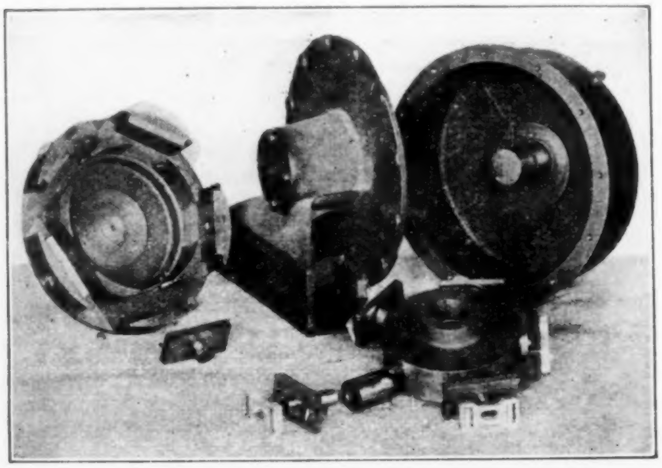

Fig. 8 Disassembled View of Detached Rotary Motor Controlled by Oilgear Hydraulic Transmission

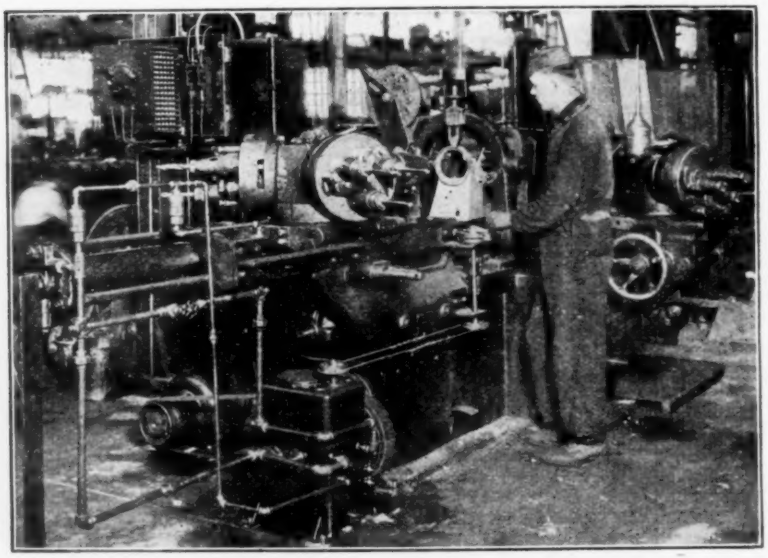

Fig. 9 Special Turret Machine Equipped with Hydraulic Feed

5 to 7, inclusive, connected to a feeding motor for driving a machine-tool carriage or ram, either a direct-acting pushing cylinder as shown in Fig. 5 or a rotary motor as shown in Fig. 8.

In either case the motor is located with reference to the feed mechanism to be driven. The direct-acting feeding cylinder is always to be preferred when it can be used, because of its well-nigh perfect steadiness of operation and the great range of feeding and rapid traverse speeds which can be satisfactorily obtained. For instance, a 4-in. piston may be driven by the variable-stroke pump of Fig. 6 at feeds of $^{3}/_{8}$ in. per min. or less, while the rapid-traverse pump (K, Fig. 7) will move the piston at 14 or 15 ft. per min.—a speed ratio of about 450 to 1.

Assuming that a rotary feeding motor can be driven at 1000 r.p.m. for rapid traverse movements, the same speed ratio would require it to turn steadily at $2^{1}/_{4}$ r.p.m. when pulling a heavy feeding cut. This range is more than such a motor will satisfactorily handle. In addition, the efficiency of the direct-acting cylinder is much greater, and mechanism is saved.

The application of Oilgear feed has so far been carried out only on already existing machine tools, but several feed installations have been in continuous operation on high-production work for periods of from one to two years, and in no case has there been the slightest wear of the plungers or other working parts.

Fig. 9 shows the application to a special turret machine producing gas-engine beds.

Hydraulic-Press Applications

While the above-described control pump was primarily designed for a machine-tool feed, its properties have proved highly desirable in hydraulic-press work. Such a press, of 25 tons capacity, is shown in Fig. 10. It may be driven from any constant-speed source of power, such as lineshaft or constant-speed motor. Rapid

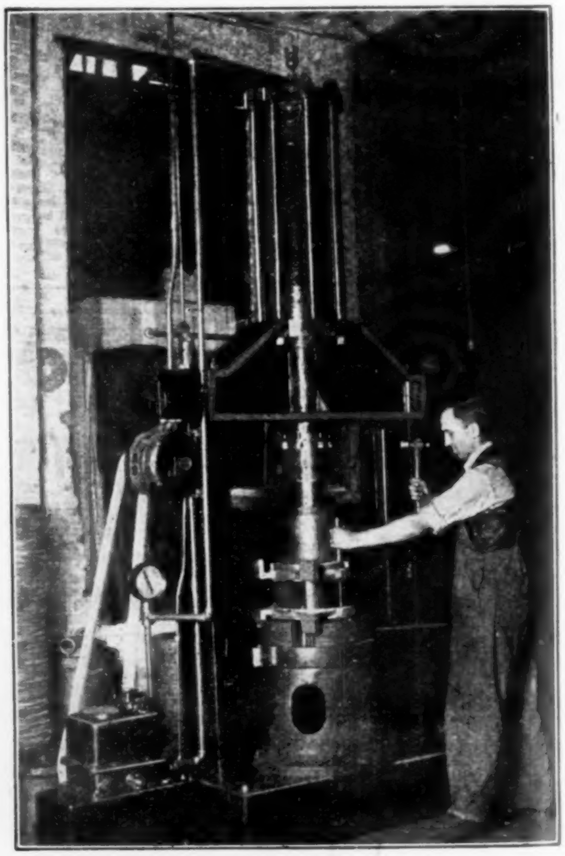

Fig. 10 Hydraulic Press Operated by Oilgear Pump

traverse of the ram is furnished, at a speed of about 40 in. per min. downward and 60 in. per min. upward. While pressing, any speed of the ram up to about 7 in. per min. is available, and the control is so delicate that when the ram is absolutely stalled against a shoulder the pressure gage can be maintained at any desired point. Many other presses driven by Oilgear pumps and having similar operating characteristics are in use.

Broaching. An important application of the direct-acting cylinder is the ordinary broaching process. A horizontal hydraulic broaching machine shown in the complete paper is driven by a 10-15 hp. Oilgear pump. The pulling cylinder is 6 in. in bore, and is double-acting. The piston rod is attached directly to the sliding head which pulls the broach. The reversing tappets are engaged by the sliding crosshead at either end of its stroke as in a planer, and are connected to the pilot valve of the hydraulic reverse gear in the Oilgear pump. The connection to the pilot valve is from a vertical rockshaft whose angle of swing is limited by two adjusting screws, thus controlling the stroke of the pump separately during the pulling and return strokes of the broaching machine and permitting any desired speed up to a maximum of 360 in. (30 ft.) per min.

In the ordinary method of pulling the broaches by screws the usual speed is less than 5 ft. per min., and the hydraulic broaching machine therefore gives a great increase of output, probably averaging 100 per cent. The comparison between the screw and the hydraulic broaching methods may be approximately given by the statement that on the average one hydraulic machine and operator will replace two screw machines and two operators and require about the same amount of power.

Vertical Push-Broaching Presses. The vertical semi-automatic manufacturing press is a further illustration of the adaptability of hydraulic drive to produce machine tools exactly adapted for the work in hand. This press is equipped with an automatic stroke-control gear and is driven by a 5-hp. Oilgear variable-stroke pump known as the Type W. The length of stroke, and hence the speed of the ram, is set by the location of the adjustable tappets. The typical working cycle consists of a downward rapid-traverse stroke at any desired speed through any desired distance, followed by a working stroke at a different speed through any additional distance, automatically reversing and returning to the top position at the completion of the stroke. The ram may be set either for full stroke or for as short a stroke as will be sufficient for the work in hand, and this stroke may be taken near the upper position of the ram or at any other point within its total range. If the entire downward stroke is a working stroke, the rapid approach stroke may be omitted, the entire stroke being set at the required working speed.

Straightening Presses. Equipped in this way this press is available for various vertical broaching and forcing operations. When desired the press may be operated by the QC control pump shown in Fig. 5 and used as a sensitive straightening press. When so equipped, the operator can move the ram by thousandths of an inch if desired, and apply pressure accurately by the pressure gage.

Hydraulic Riveting. A third and even more striking application of such a press is its use as a hydraulic riveter. In this case it is equipped with a pump known as the Type WE, which differs from the Type W only in the method of controlling the stroke. The Type WE pump, is so arranged as to always pump a maximum quantity of oil when the pressure is below a certain point, say, 900 lb. per sq. in. As the pressure rises the pump automatically reduces its rate of delivery until at a predetermined maximum of, say, 1000 lb. per sq. in. the delivery ceases altogether except for the small quantity required to maintain the maximum pressure in the delivery pipe.

When this kind of a control is used the pump is connected to the riveter through a four-way piston valve. The operator handles the press or riveter by operating the piston valve and the pump functions exactly like an accumulator, except that the oil pump only develops sufficient pressure at any instant to overcome the actual resistance of the ram. When the ram is stalled against a rivet or other dead stop the power consumption ceases and the pump runs idle except with sufficient delivery to maintain the pressure.

Such a plant in effect is a portable accumulator plant operating at an overall efficiency of 85 per cent or more, and adapted to be driven from any convenient source of power such as a lineshaft or small electric motor. It requires but from one-eighth to one-sixth the power needed to operate an air riveter, and its use will frequently avoid increasing air-compressor capacity.

Rotary Machine-Tool Drive

The most obvious field in the machine shop for the variable-speed hydraulic transmission is in the main drives of the machine tools themselves. On the other hand, it is the most difficult part of the field to develop, because of the required changes in designs and patterns, but still more because of the high state of development attained by the direct-current electric motor on variable-speed work. Nevertheless, the hydraulic variable-speed drive has several important advantages even over the best electric variable-speed drive, including the outstanding advantages of offering a means of obtaining a perfect speed variation from an alternating-current motor. These may be stated as follows:

a More perfect control of speed
b No coasting when the power is shut off
c No peak loads drawn from the line, and no heating of the hydraulic drive
d Low maintenance and minimum of attention required.

In equipping the average machine tool with hydraulic drive, some saving will be obtained from the omission of a number of gear changes, etc., now required, and a further saving from the fact that all the machines of a certain kind and size will be identical, no matter what the source of power used to drive them. That is, a lathe headstock equipped for hydraulic drive would have a single receiving pulley or shaft to be driven at, say, 600 or 800 r.p.m. It might be driven either from a lineshaft, from an alternating-current motor, from a direct-current motor, or from a gas engine. If such a machine tool came on the second-hand market it would be equally available for installation anywhere.

The mechanism and advantages of hydraulic feed have already been touched upon. Its introduction into new machine tools will be somewhat modified if it is carried out in connection with the use of hydraulic drive for the same tools. In any case the use of the hydraulic method on any machine tool opens the way for the use of hydraulically operated friction clutches to effect gear changes, and also for the replacement of the ordinary air chuck by hydraulically operated chucks. This has already been done experimentally and with complete success. The hydraulic chuck has advantages over the air chuck in consuming much less power, having no tendency to dry out the cup leathers and thus fail to operate, and in moving only sufficiently to release and grip the work instead of making the entire stroke whenever operated.

All of the applications described in the preceding paragraphs have been actually made, and most of them have been in use a sufficient length of time to prove that they are successful practically. Many important fields are not yet touched, although the properties of the Oilgear mechanism seem perfectly adapted to their requirements. Among these are planer drive and other reciprocating movements, cranes and hoists, and elevators.

Discussion

In opening the discussion, J. J. Crain[1] said that the concern with which he was connected had been manufacturing hydraulic transmissions of the same general nature as the one described by the author for about 15 years, and had built about 2000 of them, most of which had gone to the United States Navy, where they were used for turning turrets, elevating guns, hoisting ammunition, etc. All of the present capital ships used this device exclusively for these purposes. The author was very modest in his claims of the performance of this type of apparatus. Before hydraulic transmissions were used in the Navy, most of the high-grade control was effected by the Ward-Leonard system and while this gave very fine control, it was not comparable to what could be obtained from a variable-stroke pump.

His company's associates in England had built about the same number of machines, but had gone more exclusively into commercial work. The real problem in wide commercial application was the financial one. It was an expensive machine. It had to be built accurately and of high-grade material, and to convince a man to pay for it it was necessary to show him that it would produce annual savings which would warrant the investment.

The author seemed to have attacked the most difficult part of the problem first, and with no little success. The control given was without question practically perfect and there was practically no discussion on that point; rather, it was what price could the man who used it afford to pay?

The noise increased with the larger units, that being one of the limitations to the size that could be employed.

Mr. Crain said that he had not found very much difference in efficiency between the large units and the small units, the efficiency lying between 82 and 85 per cent at full speed and full power.

Charles M. Manly,[2] referring to an efficiency curve given in the complete paper, said that it was a curve of efficiency at constant torque and not at constant horsepower. In tests which he had made and presented to the Society[3] in 1911, the efficiency at constant horsepower had been shown to be greater than the effi-

(*Continued on page 245*)

[1] Manager, Waterbury Tool Co., Waterbury, Conn. Mem. A.S.M.E.
[2] Manly & Veal, Consulting Engineers, New York City. Mem. A.S.M.E.
[3] Variable-Speed Power Transmission, George H. Barrus and C. M. Manly, Trans. A.S.M.E., vol. 33, p. 851.

Safety Engineering in the Compression of Gases

By A. D. RISTEEN,[1] HARTFORD, CONN.

The purpose of this paper is to outline a few of the chief hazards that are associated with the compression of some of the gases in common use in industry. Among the gases considered are air, oxygen, nitrogen, argon, carbon dioxide, hydrogen, acetylene, ammonia, and chlorine. The preparation, utilization, storage, or transportation of the gases, except when some of these items may happen to have an important bearing upon the actual operation of compression, are not discussed. As to the mechanical strength of the apparatus that is used to effect the compression, the paper deals only with the things that are likely to happen even when the apparatus itself is strong enough to withstand the stresses that are thrown upon it in the course of its normal operation.

COMPRESSED air, the first of the gases to be considered, is used for many different purposes and at many different pressures, but so far as safety is concerned it is sufficient to distinguish two main problems. First, we have to deal with cases in which the ultimate pressure desired does not exceed, say, 200 lb. per sq. in.; and second, the special case in which the compression must be pushed to perhaps 3000 lb. per sq. in. for the production of liquid air.

When the ultimate pressure is not greater than 75 lb., the compression is usually effected in a single operation; but it would seem better to adopt two-stage compression for pressures approaching or exceeding this limit. Three stages, at least, should be used for pressures in the neighborhood of 1000 lb., and to push the pressure from this point up to 3000 lb. a fourth stage or operation should be employed. It is advisable to use long-stroke compressors, with low piston speed, not only to avoid unnecessarily high temperatures, but also because less lubrication is needed.

Every now and then there is an explosion in connection with an air compressor, and the results are sometimes quite serious. Most of these explosions are probably associated more or less directly with the lubrication of the compressor cylinders, and with inefficient cooling of the air that is undergoing compression. In an ordinary compressor the temperature of the air rises considerably during the compression stroke, and it is of the utmost importance to keep this rise of temperature within reasonable bounds. A good deal can be accomplished in this direction by surrounding the cylinders with effective water jackets. The cooling water in these jackets should circulate actively and copiously, and thermometers should be provided to show the temperature at which the water is entering and leaving. To guard against stoppage of the flow of cooling water from any cause, it is also important to provide some positive means for showing that the circulation is free and plentiful at all times. It is advisable, for example, to have the discharge from each jacket located where it is plainly visible to the men working about the room.

It is not sufficient, however, to provide the compressor cylinders with water jackets. The air should be passed through a special cooler immediately after leaving the compressor and its temperature brought down near that of the surrounding atmosphere as quickly as practicable. Moreover, if the compression is effected in two or more stages, an intercooler between each stage and the next one should be provided, so that every cylinder will be supplied with air at a moderate and reasonable initial temperature. The use of efficient cooling devices tends not only to insure safety but also to lessen the cost of compression, because cooling reduces the pressure as well as the temperature.

In compressing air to moderate pressures the cylinders are lubricated with oil. Oil may also be used in the first stages of high-compression apparatus, though water is preferable for the final stages. The composition and physical characteristics of the oil should be carefully considered.

As a general rule, too much oil is used in lubricating the cylinders of air compressors. It takes considerable experience to determine just the right amount, and what is right with one machine or one grade of oil may not be right with another. In a general way, however, it is safe to say that the lubrication is ample if the cylinder walls are always coated with a slight film of oil. It is often assumed that the explosions that sometimes occur in connection with air compressors are due to the ignition of oil vapor or of oily mist. It would be hard to justify this theory. Whenever a compressor cylinder discharges the air that it contains, the oily vapor or spray that is present is discharged at the same time, and it is hard to see how a quantity sufficient to produce a serious explosion could accumulate. Moreover there are comparatively few explosions in which the initial break is in the compressor cylinder. More commonly the first rupture occurs in the discharge pipe or the receiver, and trouble at these points can be minimized, and perhaps entirely avoided, by (1) quick and efficient cooling of the air, (2) thorough drainage of the oil that tends to collect in the piping and receiver, and (3) careful attention to the discharge valves on the compression apparatus. Considerable quantities of oil are likely to accumulate in the discharge pipe and the receiver, and if either of these bursts, a large amount of hot oily spray is likely to be discharged into the air, and this may take fire either spontaneously or from some external source, with a bad oil-vapor explosion as the result. Suitable traps and drains must be provided to prevent the accumulation of the oil, and in this way the oil hazard can be largely or wholly removed.

If the exhaust valves of the compression apparatus are leaky, then upon the return stroke more or less of the compressed and heated air will find its way back into the cylinder, so that the temperature of the air in the cylinder upon the next compression stroke will reach a much higher point than the designer of the apparatus intended. Carbonaceous deposits, produced by the carbonization of the lubricating oil, often collect in considerable quantity in and around the exhaust valves, and these may sometimes take fire, or they may prevent the valves from closing tightly. Exhaust valves should be carefully watched for such deposits, and should be kept as clean and tight as possible.

Oxygen

Oxygen is compressed and shipped in vast quantities and is used in the greatest imaginable variety of ways. Before subjecting it to compression it is essential to know that the oxygen is pure, and especially to know that it is not contaminated with any other substance with which it could combine, either during compression or in the course of subsequent handling. For example, if it is prepared by the fractional distillation of liquid air, it is important to know that the air that was subjected to liquefaction was free from smoke particles, from organic dust of any kind, and from coal gas, acetylene, and every other substance of an oxidizable nature. If the oxygen was produced by the electrolysis of water, it is equally essential to know that it is not contaminated by hydrogen. It is easy enough to insure purity in all these respects, but the importance of not neglecting the necessary testing, and the purification operations when these prove to be needed, cannot be over emphasized.

In handling compressed oxygen it is supremely important to prevent the gas from coming in contact with oil or grease either during the compression or while the oxygen is being stored, transported, or utilized; and in view of the fact that many exceedingly serious accidents have occurred in consequence of neglecting this principle, the author wishes to emphasize the hazard just as strongly as he can. When a man talks with appropriate earnestness about the danger of allowing compressed oxygen to come in contact with oil, grease, or other combustible organic materials, it is easy for the uninitiated to believe that he is inspired by unreasonable timidity, but this is not the case. Those who have followed the history of oxygen compression are well aware that many accidents have occurred under conditions that thoughtful and experienced men

[1] Director of Technical Research and Safety Publication Work, Travelers Insurance Company and Travelers Indemnity Company. Mem. A.S.M.E.

Presented at a joint session of the American Society of Safety Engineers and the A.S.M.E. Safety Codes Committee at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. All papers are subject to revision.

would not have considered to be at all likely to produce trouble. The use of even a slight trace of oil in the compressor is exceedingly hazardous, not only on account of the danger of immediate combustion, but also because some part of the oil is bound to go over with the compressed gas; this will collect in the tanks or cylinders used for storage purposes, and sooner or later serious results will surely follow. It is probably safe to say that most of the accidents that have occurred in connection with compressed oxygen have been due to lack of appreciation of the importance of this thing. The trouble is not confined, of course, to the mere combustion of the oil itself. The metal parts of the apparatus will burn freely as soon as they become sufficiently heated by the ignition of the oil or grease, and the consequences may transcend imagination.

The only absolutely safe thing to use for the lubrication of an oxygen-compressor cylinder is *pure water*, which should be introduced into the suction pipe in the form of a visible spray. Under special conditions it may happen that pure water is not sufficient, and in a case of this kind it may be permissible to dissolve a little soap in it; but soap should not be used when water will suffice, and it should *never* be used without first proving, by actual analysis, that the particular supply to be dissolved contains no uncombined (or unsaponified) oil or grease. The packings used on the compressor must also be entirely free from oil, grease, or graphite. Packings made for this express purpose should always be employed.

The compression of oxygen is often, and perhaps usually, carried up to about 2000 lb. per sq. in., and it should be effected in not less than three stages. In speaking of the compression of air, the importance of prompt and effective *cooling* has been emphasized, and much greater attention should be paid to this point when compressing oxygen. The pump cylinders should be water-jacketed, the circulation of water should be positive and abundant, and the apparatus should be so designed that cessation or diminution of the flow of jacket water cannot escape immediate notice. The gas should also be passed through effective intercoolers between each compression stage and the one next following; and *immediately* after leaving the last cylinder of the compressor it must be passed through a final cooler, and brought down to the temperature of the surrounding atmosphere.

It is exceedingly unwise to introduce oxygen into receivers of any kind, unless these receivers are known to be perfectly clean internally and wholly free from oil, grease, and other combustible substances. The author would like to see some kind of a regulation established forbidding every oxygen manufacturer to fill any tanks other than his own. This principle is already followed, it is believed, by some and perhaps all of the larger manufacturers, but unfortunately it is not followed by everybody. Moreover, oxygen should never be compressed into containers that have been used for other gases. When a cylinder or tank has been previously used for holding compressed gases other than oxygen, there is likely to be an accumulation of oil upon its inner surface, because oil may have been used for cylinder lubrication in connection with the compression of these other gases, and it is hard to remove such oil thoroughly enough to make the tank safe for oxygen.

NITROGEN

Nitrogen gas has a far narrower range of application in the arts than oxygen, yet it is compressed and shipped to a certain extent. This gas was used during the war in the recoil cylinders of certain types of guns, and it has also been used to a limited extent for blanketing combustible liquids and for supplying a neutral atmosphere for other purposes, where the presence of a gas with active chemical properties would be objectionable. At the present time compressed nitrogen is supplied mainly, it is believed, for filling incandescent electric lamps, and in connection with the manufacture of automobile tires.

Nitrogen forms four-fifths of the bulk of the atmosphere, and as it is nearly inert in its elemental form the precautions that are recommended in connection with the compression of air are also adequate when compressing nitrogen. It should be noted, however, that the nitrogen that is used for filling electric-lamp bulbs must be free from hydrocarbons, and this means that when the gas is to be employed for this purpose it is not permissible to use oil for lubricating the compressor cylinders. Water should be used instead. For certain reasons, in fact, it is best to avoid oil altogether and to lubricate with water in all cases. This does not mean, however, that there is any danger of direct chemical action between the oil and the nitrogen.

Nitrogen is usually compressed to about 2000 lb. per sq. in., and the compression should be performed in three stages.

ARGON

Argon, like helium and neon, has no chemical properties whatsoever. It forms no compounds and is absolutely inert under all circumstances. Hence it cannot produce fires or explosions. Moreover, it is not poisonous. It is obtained from liquid air, however, and when first separated from the air it is mingled with a large proportion of oxygen—the mixture usually containing about 65 per cent of oxygen and 35 of argon. When it is handled in this state it must therefore be treated like oxygen; but after it has been freed from the admixed oxygen it may be compressed under the same conditions as nitrogen. For shipment it is usually compressed to a pressure of 1800 or 2000 lb. per sq. in., in three stages.

CARBON DIOXIDE

This gas is compressed in large quantities and is used for the most varied purposes. As is well known, it will neither burn nor support combustion, and the author does not recall any chemical dangers which are likely to be encountered in its compression. It is not especially poisonous, but when a considerable quantity of it is mixed with the air that the workmen have to breathe, it replaces the life-giving oxygen to a corresponding extent, and this may mean that the air is not fit for respiration. The effects of carbon dioxide in this way are sometimes very insidious, especially as the pure gas is without odor or color; and it is therefore evident that special care should be taken to provide free and copious ventilation at all times in buildings in which carbon dioxide may escape, by leakage or otherwise.

HYDROGEN

In connection with hydrogen and most of the other combustible gases (such as propane), the chief danger associated with compression probably consists in the likelihood of fire or explosion in case of leakage or of contamination with oxygen. If the hydrogen is prepared by the electrolysis of water, there is always a possibility that oxygen may be present in it in small amounts, and the gas should be carefully tested, either at short intervals or preferably by some continuous process, to make sure that the oxygen content is always well below 5 per cent. Small amounts of oxygen can be removed by passing the gas over palladium pumice, which causes the oxygen to combine in a quiet way with an equivalent quantity of hydrogen.

The danger of admixture with air or oxygen is not in any way restricted to hydrogen prepared by the electrolytic process. Air may leak into the suction pipe leading to the compressor, whatever the source of the hydrogen; and this, in fact, is one of the hazards that must be watched most carefully in handling this gas. The suction line may be fitted with test cocks or drainage valves, and when this is the case it is exceedingly important to keep them closed except when they are opened for legitimate purposes and at proper times. It is believed, moreover, that all such cocks and valves should be kept locked, and that the keys should be in the keeping of some designated, responsible man. To guard further against trouble from this source and from accidental leakage, the whole operation should be conducted so that there will be a positive pressure in the suction line *at all times*. Care should also be taken that the gasometer does not stick and that its seams are kept absolutely tight.

It is important to consider the possibility of leakage outward as well as inward. Hydrogen is usually compressed to a pressure of 1800 or 2000 lb. per sq. in., and at high pressures a good deal of gas will escape through a small hole in a short time. If hydrogen gets out into the room and is allowed to collect there, a serious explosion is likely to result. Every discoverable source of ignition should be carefully considered and safeguarded. Rigorous measures should be taken to prevent the men from smoking or using matches or other sources of open flame. The artificial lighting should be provided by incandescent electric lamps enclosed in

vapor-proof globes, and it is safest to mount these lamps outside of the windows, locating them so that they shine through the window glass into the interior. All electric wires should be run in closed conduits, and the fuses should invariably be outside of the region of possible danger. This applies also to switches, unless they are of a type which cannot produce an arc or flame. No electric motors should be used in rooms where hydrogen, propane, or any other combustible gas may escape in quantity, unless these motors are specially designed for this particular application, or else thoroughly ventilated by fresh, pure air from the outside of the building. And these various precautions should be observed not only where the hydrogen is compressed, but also wherever it is stored—either in gasometers or in cylinders. Finally, it is exceedingly important to provide abundant ventilation at all times to lessen the chance of fire or explosion in case the other precautions prove inadequate.

Left-handed threads are extensively used for fittings that are to be employed in connection with hydrogen, the purpose being to prevent the accidental connection of hydrogen lines or cylinders with pipe lines conveying other gases, and especially with those containing air or oxygen. It is to be regretted that in many plants the purpose of the left-handed thread is deliberately defeated by making use of devices known as "adapters." These are in effect right-and-left couplings, which are provided for the express purpose of making it possible to attach a cylinder with a left-handed thread to a pipe line having a right-handed thread.

Acetylene

No person should ever undertake to compress acetylene unless he fully understands the properties of the gas and knows exactly what it is permissible to do and what he must carefully avoid doing. This is not meant to imply that the compression of acetylene is a dangerous operation when it is performed under the direction of a properly qualified man, but is intended as a special note of warning to those who are tempted to compress the gas without first understanding its properties. Acetylene is an endothermic compound, and this formidable word has frightened a great many persons who do not understand its exact significance. It merely means that heat is absorbed (instead of emitted) when acetylene is formed from its constituent elements. Now it is well known that substances of this kind are likely to be more or less unstable under certain conditions, and acetylene is no exception to the rule; but the conditions under which the instability exists are well known, and it is only the man who does not understand them, or who is pleased to ignore them, who gets into trouble.

In compressing acetylene it is important to keep the temperature of the gas as low as practicable at all times. Expressed in more definite language, this means that the compression must be effected by a pump moving with a very low piston speed, and that it must be performed in not less than three stages. It also means that the pump cylinders should be kept as cool as practicable by a plentiful supply of water drawn from the coolest available source. The author cannot state any definite temperature at which the gas may become instable during the act of compression, and in fact it is doubted if anyone knows what that temperature might be. The only thing that we can say is, that if the operation is conducted as here described, there is apparently no danger from instability. It certainly is not safe, however, to *store* acetylene in otherwise empty cylinders, at the pressure at which the compressor pumps deliver it. There appears to be a time element involved here, and it is probable that a certain amount of polymerization gradually occurs in strongly compressed acetylene (except when it is stored in special receptacles containing suitable porous material charged with liquid acetone), so that in the course of time small quantities of other more sensitive substances are formed, which serve to lessen the stability. At all events, we know that acetylene should not be stored in an otherwise empty tank at a pressure of more than 15 lb. per sq. in. greater than the normal pressure of the atmosphere, and yet we know that explosions in the compression pumps, and in the delivery pipes leading from them, are exceedingly rare if the compression is effected as here described. In fact, the author knows of only one case of the sort, and this was precipitated by the breakage of a steel valve spring, which quite possibly struck a spark and thereby caused the development of a high temperature within the tiny space filled by the spark itself.

Oil may be used for lubricating the cylinders of acetylene-compressing pumps, but it should afterward be removed by passing the compressed gas through suitable separators as it leaves the compressor. And immediately upon leaving the oil separator the acetylene should be passed into the storage cylinders, which, as previously stated, must be filled with a special porous, solid material thoroughly impregnated by liquid acetone, which dissolves the acetylene and holds it safely in solution.

It should not be necessary to issue a warning against mixing other gases with acetylene in the storage tanks, or using acetylene storage cylinders for any purpose whatsoever, except for holding pure, unmixed acetylene (in addition to the safety filling).

Ethyl Chloride, Methyl Chloride, and Propane

These gases are used to some extent as refrigerating agents, but it does not appear desirable to discuss their compression in any detail. They are all inflammable, and hence must be handled with due regard to the possibility of fire and explosion. Propane (like butane) does not require lubrication in the compressor cylinder, because the substance itself affords sufficient lubrication. It is necessary, however, to lubricate the stuffing box of the compressor. There has been some difficulty in connection with the lubrication of cylinders in which the chlorides of ethyl and methyl are compressed. Castor oil has been used, but it is not very satisfactory. Glycerine is said to be much better, though still far from ideal. It is said that a new lubricant, consisting of ethylene and propylene glycol mixed with deflocculated graphite, and specially adapted to ethyl and methyl chlorides, and to propane, is now available.

In handling these and all other inflammable gases the various special precautions mentioned in connection with hydrogen should also be observed.

Ammonia

Liquefied ammonia is used in immense quantities in connection with refrigerating machinery, and although it perhaps cannot be called a poisonous gas in the narrow sense of the term, it is easily capable of producing unconsciousness and death when it is present in the air in any considerable amount. Similar remarks apply to sulphur dioxide gas, which is also used in refrigerating machinery as well as in various other applications. Both of these gases are freely soluble in water, and it is recommended that in all rooms in which they are handled an overhead sprinkling system be provided that is capable of discharging large volumes of water into the room in case of the accidental liberation of excessive quantities of either gas. A considerable part of the liberated gas would be absorbed by the down-rushing water, and even though the air might not be rendered respirable in this way, something would surely be gained in the way of checking the spread of the gas to other rooms. It is certainly important to provide free and abundant ventilation in and adequate means for quick exit from any room in which either gas may suddenly be liberated in quantity. These simple and evident precautions are often wholly disregarded in laying out plants in which ammonia or sulphur dioxide are handled or used.

Ammonia gas is usually pronounced incombustible and incapable of being exploded when mixed with atmospheric air, but research and experience have shown that this is not altogether true. A mixture of ammonia gas and air can be exploded if it contains from 16 to 27 per cent of ammonia, and some of the explosions that have occurred in our big refrigerating plants subsequent to the liberation of considerable quantities of ammonia may possibly have been due to the ignition of mixtures of this kind by means of the arc lamps which have in most cases been in use in rooms where these accidents have occurred.

In compressing ammonia another source of danger must be carefully considered. If the ammonia gas that is undergoing compression is allowed to become unduly hot, the ammonia that comes away from the compressor will contain a notable quantity of combustible gas that is distinctly different from ammonia. This may come in some measure from the decomposition of the oil that is used in lubricating the compressor cylinder, but it also contains free hydrogen, which appears to be produced by the actual dissociation (or breaking up) of the ammonia into its component gases, nitrogen and hydrogen. We used to think of ammonia as being exceedingly stable and incapable of dissociating in this way

to any sensible extent, but we now know that such is not the case. Dissociation occurs even at moderate temperatures, and it is likely to become quite significant when the temperature is high. To guard against the development of combustible decomposition products from the lubricating oil, it is important to use a special kind of oil which experience has proved to be well adapted to work of this kind; and to keep the dissociation of the ammonia itself within as low a limit as practicable, it is important to keep the gas as cool as possible by methods already outlined in connection with other gases. Finally, to guard against vapor explosions in case ammonia gas should escape into the air in considerable quantity, it is important to take the same precautions against ignition sources that have been suggested in connection with hydrogen.

Chlorine

Liquid chlorine is now shipped in vast quantities, for use in connection with bleaching, and for many other purposes. It has been greatly feared by the general public ever since it was used as a military gas in the war, but in time of peace this fear is hardly justifiable. Chlorine is exceedingly irritating, and it can produce unconsciousness and even death if inhaled in any considerable quantity. It has been well described as an "honest gas," however, the phrase meaning that it has no treacherous qualities. We always know exactly what it will do. In a manufacturing plant, or in a plant in which chlorine is compressed, immediate relief can be had, in case of leakage, by merely leaning out of an open window or going out of the room. It is not desired to underrate the dangers associated with handling liquid chlorine, yet it is only fair to say that they have been largely exaggerated. If a cylinder or a tank containing liquid chlorine bursts or springs a leak, for example, the evaporation that takes place rapidly chills the liquid still remaining in the tank, so that the evolution of gas is quickly and automatically checked, though not wholly stopped. Abundant ventilation is important in a chlorine plant and special attention should be given to the exits, so that if the gas becomes accidentally liberated in considerable quantity the men can pass out into the open air as quickly and directly as possible.

Chlorine must be carefully dried before compression, because if it is moist it will corrode the pump, piping, and tanks. Perfectly dry chlorine, on the other hand, has practically no action upon iron. The drying is effected by passing the gas through towers filled with pumice wet with concentrated sulphuric acid. In liquefying the gas by means of a compression pump the main difficulty to be overcome is keeping the piston gastight in the compression cylinder. A special form of packing is required for this purpose, and it should be designed and installed by some person who understands the necessities thoroughly. In one compression plant visited by the author there are seven sealing rings on the piston—six soft and hard packings of special chlorine-resisting material being used, while the middle space is filled with strong sulphuric acid which not only completes the seal but also serves to lubricate the cylinder. The pressure required in chlorine tanks is not high, as the vapor pressure of liquid chlorine is only about 120 lb. per sq. in., at 70 deg. fahr.; and even when a chlorine tank has been left standing in the sun for a considerable time, the pressure will hardly ever creep up as high as 160 lb. per sq. in. In liquefying this gas the compression is therefore performed in one operation. Tanks and other apparatus used in handling or storing chlorine must be clean and free from all other substances—solid, liquid, or gaseous—with which the chlorine might combine.

In conclusion, there is one general suggestion relating to safety-valves which applies to all gases. Some engineers discourage the use of safety valves on gas-compressing apparatus, partly because the escape of gas through them is a source of waste, and partly because it creates a special hazard, when the gases are toxic or inflammable. But it is not necessary to liberate the escaping gas at or near the work place, and it can often be discharged where it will not constitute a hazard to property. Hence this objection is not altogether well founded. In fact, relief valves should be provided in all cases, and the discharge problem should be considered on its own merits in each individual plant. It is often, and perhaps usually, practicable to have the discharge delivered back into the piping system on the low-pressure side of the compressor, so that there is no economic loss, and no hazard to the men nor to property.

Discussion of Hydraulic-Transmission

(*Continued from page 241*)

ciency at constant torque, so that the author's curve would be better at constant horsepower. This was a condition which was of great interest in many applications where it was possible to get increased torque with decreased speed and full horsepower within a reasonable range.

Of course a full horsepower output could not be obtained throughout an infinite speed range. If one ordered a machine for a 10 to 1 range at a constant capacity and needed only a 5 to 1 range, he would have to buy a machine of twice the capacity actually necessary.

In closing the discussion the author said that Mr. Manly had raised an important question in his comment on the increased efficiency shown in his tests at constant horsepower. He did not know whether a similar condition existed in the Oilgear, as it depended on the relative values of the different components of the total loss, such as mechanical and hydraulic friction.

Of greater practical importance was the method of obtaining the constant horsepower output. If this was done by varying the pump stroke, the working pressure would increase as the pump stroke was reduced. This amounted to underrating the machine in order to call it a constant-horsepower machine. If it was to operate at half-stroke with 1000 lb. per sq. in. pressure and at full stroke with 500 lb. per sq. in. to give constant horsepower, it might as well carry doubled rating at full stroke, unless limited by bearing pressures or speeds, or by overheating.

Regarding Mr. Crain's remarks on the subject of costly construction, the author believed that the simple plane and cylindrical parts used in the Oilgear would lead to a reduction of cost. This prospect had led him to believe that the machine-tool field and similar fields should be considered open for hydraulic-transmission methods.

The variable-stroke oil pump as a power transmitter had the following outstanding characteristics:

a Complete speed control of the driven motor
b Transmission of maximum torques to the driven motor without requiring large power input
c Automatic regulation of power input at all speeds to the amount actually required to deliver the power output
d Perfect lubrication and consequent extreme durability
e Negligible idling load
f Ability to operate as a disengaging clutch by setting the pump at zero stroke.

No other power transmitter combined these properties, and very few power transmitters had any of them. The direct-current electric motor, however, had been highly developed as a variable-speed drive, and had a large field appropriate to it in which the hydraulic drive would not compete.

It was very important to get a clear idea of the essential distinction between hydraulic variable-speed drive as herein presented and the ordinary types of hydraulic machinery. Every Oilgear transmission comprised a pump, a motor, and an oil circuit connecting them. The volume of the liquid flow and hence the speed of the motor was determined only by the pump displacement. The pressure of the working fluid was determined only by the resistance encountered by the motor, and rose and fell with the load to be driven. There were no throttle valves and no reduction of pressure in the working liquid except when it passed through the motor. Then the pressure instantly fell to exhaust pressure, the drop being just sufficient to compel the motor to turn against the mechanical resistance to be overcome, because the high pressure was determined and caused by this same resistance.

The ordinary hydraulic system, including an accumulator, required a maximum pressure to be delivered by the pump whether the work needed this pressure or not, and the surplus was wasted or wiredrawn through the throttle valve between the accumulator and the working machine. This throttle valve controlled the movement of the working ram both as regarded speed and the pressure delivered. That part of the power not needed to drive the working machine at the required speed was lost, and was also used to cut the working parts and destroy the mechanism itself.

Performance Tests of Steel Belts with Compressed-Spruce Pulleys

Particulars Regarding an Investigation of the Performance of Welded and Continuous Steel Belts Running on Compressed-Spruce Pulleys Not Specially Faced

By GARTH L. YOUNG[1] AND GUIDO V. D. MARX[2]

FROM the data available it would appear that the possibility of using thin ribbons of steel in place of leather or composition belting for power transmission was conceived by Eloesser in Germany some fifteen years ago, but that at first various mechanical troubles stood in the way of extensive adoption of the new device. It was not until a few years ago that these troubles were overcome to a sufficient extent, and it is stated that at present close to 1,000,000 hp. is being transmitted by steel-belt drives.

The authors have carried out a series of tests with continuous steel belting and compressed-spruce pulleys. Heretofore the two outstanding difficulties with steel belts have been the impossibility of securing a satisfactory joint and the fact that the pulleys must be specially faced in order not to run metal on metal, which would ultimately, perhaps after a short period of satisfactory operation, give rise to excessive slip and consequent power loss. Because of this the combination of continuous belting and compressed-spruce pulleys would appear to be of particular interest.

Compressed-Spruce Pulleys

The driver and driven pulleys were 20 in. in diameter, 6 in. face,

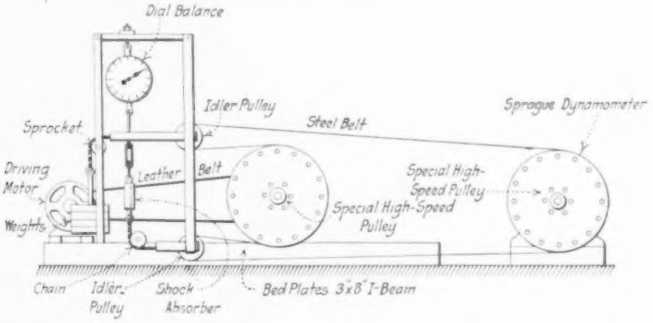

Fig. 1 Diagrammatic Sketch of Steel-Belt Testing Machine

compressed spruce, and the idlers 12 in. in diameter with 4 in. face. According to the manufacturers, "The method used in constructing these pulleys is to lay up $^3/_8$-in. thick lumber in square blocks with the alternate layers at right angles, glueing with Casco casein joint glue to whatever thickness is desired, place in a hydraulic press and compress the block perpendicular to the plane of the layers, clamp and hold for from 48 hours to a week. The clamps are then removed and the block remains compressed with practically no expansion. The block is then bored with a hole that is slightly smaller than the outside of the cast-iron hub. The machined hub is then pressed in by hydraulic pressure and the pulley turned on a mandrel in an engine lathe. The pulleys are then doweled with fir dowels, the set-screw hole (if required) bored and tapped, placed on the mandrel again, and sanded and finished." A block of this material was compressed under a pressure of 4800 tons per sq. ft., or around 68,000 lb. per sq. in. without having it fail by spreading out, and although these pulleys have never been tested to destruction by centrifugal force, it would appear from the foregoing that the wood in tension will carry up to far above ordinary driving speeds.

Belt Material and Joints

The material used for belting in these tests was what is known commercially as clock spring. It is a very high-carbon charcoal steel apparently rolled and ground to size and drawn to a dark blue temper. The belt used was 0.01 in. by 0.75 in. by 35 ft. This material may be obtained in varying widths, thicknesses, and tempers. In the 0.01-in. thickness the widths obtainable are from $^1/_2$ in. to 3 in. in tempers of light straw, dark straw, and dark blue.

First tests made to determine the tensile strength of the belt were unsatisfactory due to the fact that the material was so weakened by the scarf marks in the grips of the testing machine that the belt failed in the jaws at loads of from 225,000 to 250,000 lb. per sq. in. This difficulty was obviated by looping the belt in either grip, thus giving two thicknesses instead of one. The belt then broke cleanly between the grips at a load of 275,000 lb. per sq. in.

The previous work done with belts of this material clearly shows that a silver-soldered lap joint is by far the most satisfactory to use, hence little time was spent on the construction and test of joints of different types. A triple-riveted lap joint was made up and tested, eleven $^3/_{32}$-in. Norway tinned rivets being used. This type of joint was never run on the apparatus as it was clearly far too stiff to stand the reversals of stress over the 12-in. idler pulleys. When tested for tensile strength the rivet heads pulled out and sheared at a load of 108,000 lb. per sq. in., giving an efficiency of 39 per cent.

The silver-soldered lap joint was made as follows: The ends were squared up and scarfed to a razor edge with a file and emery cloth, the length of the scarf being about three-quarters the width of the belt. The ends were then clamped firmly in place and from $^1/_4$ in. to $^3/_8$ in. of $^3/_4$-in. by 1-in. silver solder inserted. A small quantity of borax was used as a flux. Red-hot tongs were then firmly clamped over the joint until the solder melted and flowed evenly. The joint was then allowed to slowly cool, the oxide and surplus solder being polished off.

The joint used in all the test runs, covering a period of at least 50 running hours, was later cut out of the belt and tested for tensile strength in the manner described earlier. The specimen failed at the edge of where the temper had been drawn by the tongs, and not at the joint itself. The breaking load was 153,000 lb. per sq. in., giving an efficiency of nearly 56 per cent. Apparently the joint was so constructed as to be perfectly flexible and the reverse stresses over the 12-in. idlers had not affected its strength to any appreciable degree.

Description of Apparatus Used in the Tests

The apparatus used in these tests amounted essentially to a dynamometer whereby the tensions in the tight and loose sides of the belt could be accurately measured. This was accomplished by using two idlers mounted on swinging frames as shown in Fig. 1. For measuring the slip between the driver and driven pulleys a differential counter was used which was originally built by Helmick for his tests (see Mechanical Engineering, July, 1920, p. 374). The mechanism of the differential counter is such that when the driver and driven shafts rotate equally in opposite directions there is no movement of the ring gear G, Fig. 2. The light chains shown connect the counter to the driver and driven shafts.

Due to the one-to-one ratio of the sprockets and equal diameters of the pulleys there is no motion of the ring gear unless the belt is slipping. When slip occurs the rotation of the ring gear is proportional to the difference in peripheral velocity of the two pulleys. The ratio of the ring gear to its pinion is one to twenty and the ratio of the differential gears is two to one, making the ratio from the wheel shafts to the ring pinion one to ten.

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Abridgment of a thesis submitted by the authors in June, 1922, to the Department of Mechanical Engineering and Committee on Graduate Study of Leland Stanford Jr. University in partial fulfillment of the requirements for the degree of Engineer.

The motion of the ring pinion is transferred to a specially built counter by means of a light chain and sprockets. This counter is engaged and disengaged by an electrically operated dog clutch. The unit wheel of the counter is divided into tenths so that one division of this wheel corresponds to a difference in revolutions of one one-hundredth between the driver and driven pulleys.

The revolutions of the driver pulley are taken with a Veeder counter operating in the same circuit as the differential and direct connected to the differential-wheel shaft.

Test Procedure

As the load on the weight pan and the reading of the dial balance will not give directly the tensions in the belt under test because of the angularity of the belt pull, the apparatus was properly calibrated.

The next problem was to overcome the effect of vibration, which was successfully achieved with certain modifications in the apparatus.

The object of making the tests described herewith was to obtain the relation of the coefficient of friction to the main variables of pressure, driving velocity, slip velocity, and horsepower over a practical driving range. The method followed was to maintain a constant tension in the tight side of the belt and vary the load from the point of little or no slip to a value considered excessive. This was repeated at each tension for all the available driving speeds. It was first desired to check up the work of Hampton, Leh, and Helmick with steel on cork-faced pulleys and see what variations from their results could be ascribed to the use of compressed-spruce pulleys. It was found that the two sets of tests gave results approximating each other quite closely.

The relations found to exist between the coefficient of friction u and the driver velocity V_d, slip velocity V_s, and pressure P showed that the coefficient of friction varied inversely as the 0.533 power of the pressure, inversely as the cube root of the driven velocity, and directly as the 0.174 power of the velocity of slip.

By selecting thirty representative sets of values of the variables a solution was made for the constant K. The average of the thirty gave $K = 32$. The formula in its final form therefore reads—

$$u = \frac{32 V_s^{0.174}}{P^{0.533}\, V_d^{0.333}} \qquad [1]$$

and while variations will be found in the cases of extreme values of the coefficient, this equation fits very satisfactorily over the main range of values.

As it was desired if possible to find an equation of simpler form which would fairly closely apply, a determination of the constant was made in ten representative cases when the following formula was used:

$$u = \frac{K V_s^{1/8}}{P^{1/2}\, V_d^{1/3}} \qquad [2]$$

It was found that this formula could be applied with a little more variation than the one given above, when the value of $K = 32.91$ was used.

A table was constructed to show the relative efficiencies of the various types of pulleys. The method followed was to go over the data and select, at tensions in the tight side of the belt as nearly equal as possible, readings in which the velocity of slip and driver velocity were also constant. The only variable which now could affect the coefficient of friction and the horsepower transmitted was the self-adjusting value of the tension in the loose side of the belt. The coefficient of friction of steel on compressed spruce and the corresponding horsepower is higher in each case than the values of steel with cork-faced pulleys, the average ratio (horsepowers transmitted) of wood to cork being 1.36.

As regards the relation between horsepower, velocity of slip, and tension in the tight side of the belt, the results obtained by the present investigators check very closely with those of Helmick, but in the relation involving the coefficient of friction, velocity of driver, and velocity of slip, there is considerable variation between the results obtained by the two sets of investigations, the equation derived by the authors being as given above in [1], where u is the coefficient of friction, V_s the velocity of slip, V_d the velocity of driver, and P the pressure in pounds on the pulley face. The difference between the two equations shows the relative influence of the variable factors with the two kinds of pulleys.

The next series of tests dealt with the performance of continuous belting on compressed-spruce pulleys. At the time the belts were needed it was impossible to secure belts of more than 20 to 22 ft. open length, and as the specifications from the manufacturer of the belt required 20-in. diameter pulleys when reverse stress occurred, the apparatus as heretofore used became impractical. An effort was made to devise some apparatus which would give a practical drive for the continuous belt and at the same time admit of some method of accurately determining the tensions. This apparatus is described in the unabridged thesis.

Continuous Steel Belt

The continuous steel belt tested was quite different in physical appearance from the welded belt. In appearance it was bright and highly polished, the edges being slightly rounded to reduce the hazard in handling while running, although it should be borne in mind that the safest plan is to align the apparatus so that adjustment of the running belt is unnecessary. The continuous belt is

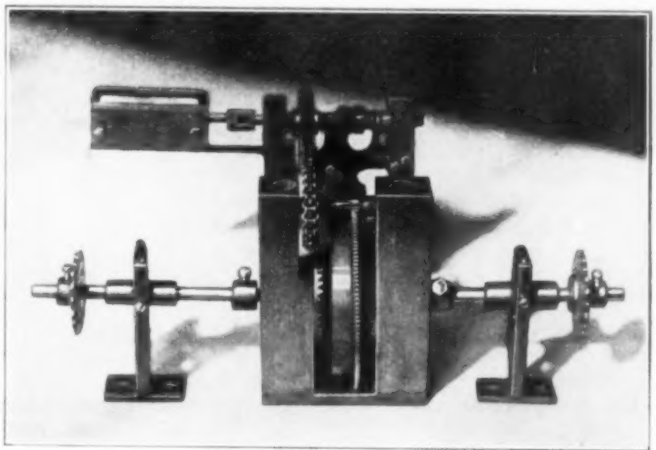


Fig. 2 Differential Counter Used for Measuring the Slip Between the Driver and Driven Pulleys

apparently not tempered to the degree of the clock spring used heretofore, and in consequence must be handled more carefully to prevent giving it permanent twists or sets. No physical tests of the material were made as it was not believed that the knowledge to be gained justified the breaking of one of the belts for this purpose.

From tests it was found that the horsepower varies as the 0.78 power of the velocity of the driver, in addition to which the following general relation was obtained:

$$\text{Horsepower} = \frac{1.51\, P^{0.81} V_d^{0.78}}{1000}$$

This formula checks out quite closely with the observed and computed data from which it was taken, with the exception that slight variation occurs at the highest velocities, when the computed value is a little high. The formula has a practical value in determining belt tensions and dimensions knowing the horsepower to be transmitted and the belt speed.

An effort was made to compute the coefficient of friction as a function of various variables in the case of a continuous belt and the values obtained proved to be extremely low, ranging from 0.192 to 0.240 in one series and from 0.198 to 0.257 in another.

The explanation of the extremely low values of u obtained in working out the data in some cases is that the floating pulley allowed the tensions in both sides of the belt to adjust themselves, and naturally at very low horsepowers, values as low as 0.0233 being recorded, the tension in the loose side of the belt became very nearly equal to that in the tight side, with the consequent result that the ratio of $(t_1 - t_c)$ to $(t_2 - t_c)$ approached unity. The logarithm of this ratio being very small, u approached zero as the limit. At just what point the values of u become reliable is difficult to determine, but it is evident from past experience that values below 0.12 or 0.13 should be disregarded.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

Vibrations in Foundations

By DR. ENG. ERNST SCHMIDT

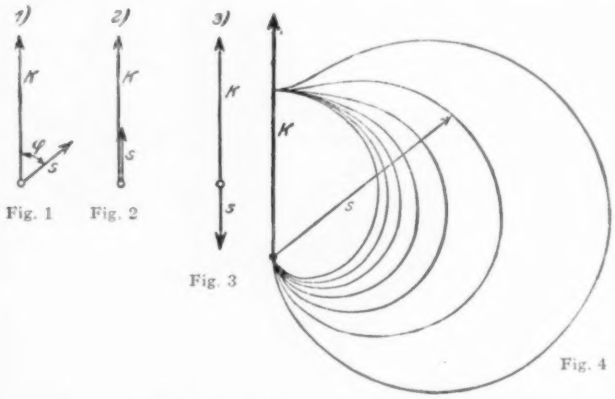
DESCRIPTION of a method of investigation based on the use of vectors. One of the features of this method is that it employs a so-called "foundation function" which expresses the motion of the foundation for every direction of freedom of oscillation in accordance with the magnitude and phase of the motion at all frequencies.

The development of vibrations in a foundation is dependent on the following elements: First, the force producing the oscillations; second, the inert mass of the machine on the foundation; third, the properties of the foundation itself; and fourth, the action of damping elements.

For theoretical purposes the investigation is limited to periodic processes which are resolved according to Fourier into harmonics, one such harmonic being taken, for example, the basic oscillation. The machine on the foundation may be considered as a rigid unit against which the oscillating masses exert periodically acting forces. The machine is assumed to rest directly or through an elastic intermediary member on a foundation which may be the ground, beams of a structure, etc.

If there is a foundation base rigidly connected to the machine, it can be considered as part thereof. The periodic forces which act on the unit assumed to be rigid are supposed to be known, and the problem is to compute therefrom the movements of the machine and foundation produced by these forces.

Let it next be assumed that the machine under consideration is



FIGS. 1 to 3 VECTORIAL REPRESENTATION OF FORCE K AND DISPLACEMENT s IN FOUNDATION VIBRATIONS

FIG. 4 FOUNDATION FUNCTION OF A FOUNDATION CONSISTING OF AN INERT MASS OSCILLATING WITH DAMPING

(The smallest curve corresponds to aperiodic damping; the larger ones belong to smaller values of damping.)

simply floating in space without connection to any kind of a foundation; in that case the inertia forces produced during the motion of the machine by the masses and moments of inertia must be in equilibrium with the forces producing vibrations. Actually, however, the machine does not float in space but rests on a foundation which holds it in a definite position. The first problem, therefore, is to investigate the influence of the "restrictions" produced thereby. Sinusoidal forces or similar moments acting on a foundation produce, when their action is restricted to only small vibrations, sinusoidal motions either of displacement or rotation. The motions generated are, however, as a rule not in phase with the forces producing them but lag somewhat behind them. If, as is usual in alternating-current engineering, the oscillating magnitudes be represented by the projection of vectors on a straight line (Fig. 1) which rotates with an angular velocity corresponding to the periodicity of the oscillations, the behavior of the foundation at any given frequency will be indicated by the vectors of the force K and the motion s, wherein the motion lags behind the force by a phase angle φ. This angle must be between 0 and 180 deg., so long as the energy from the machine is to be transferred to the foundation.

If the foundation were a perfectly elastic spring its motion would be fully in phase with the force producing it (Fig. 2). If, however, the foundation is composed of an inert, freely moving body it will move in a direction opposite to that of the force (Fig. 3). In general, φ may assume any value whatsoever; if it is acute, we approach the case of an elastic spring and the foundation may be called "springy;" if φ is obtuse, the foundation may be described as "massive."

If the frequency changes while the amplitude of the force remains

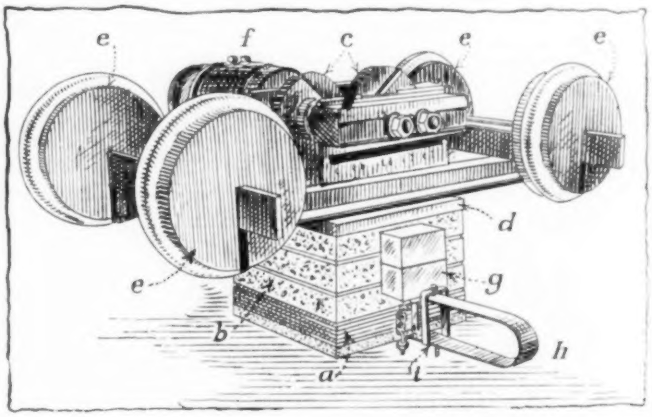


FIG. 5 DEVICE FOR INVESTIGATING THE ω FUNCTION

constant, the magnitude and direction of the vectors of the motion vary. If the end points of all the vectors of motion produced by the oscillating force at various frequencies be connected, we shall obtain a curve which may be designated as the "foundation function." If the foundation function is known for all the six freedoms of motion, the foundation may be considered as fully determined.

The energy delivered by the vibrations to the foundation may be also determined from the vector diagram as it is proportional to the area of the triangle formed by the vectors K and s.

In simple cases the "foundation function" may be computed theoretically. For example, if the foundation consists of an inert mass held in position by elastic forces and is such that its motions are damped by a friction proportional to the velocity, the foundation functions for the various magnitudes of damping will be somewhat as shown by the family of curves of Fig. 4. At low frequencies the motions and forces will be in phase. With increasing frequency the amplitude and lag of the motion increases, the amplitude reaching its maximum value at the resonance frequency where the motions lag approximately 90 deg. behind the force. From the resonance point on the amplitude begins to decrease while the angle of lag tends to approach 180 deg. At frequencies below resonance the foundation behaves in a springy manner; at frequencies above, like a mass. The foundation function may be also computed for beams yieldingly held at the ends. Since, however, as a rule, the method of holding the beam ends is not known

with sufficient exactness, such computation will not give usefully employable values. In the majority of cases occurring in actual practice, conditions are still more complicated and computation processes cannot therefore be resorted to. Here, however, the foundation function may be determined experimentally, for which purpose apparatus illustrated in the original article (Fig. 5) may be used. On the foundation is laid a plate *a* so perfectly elastic that its change of shape is at each instant proportional to the force acting upon it. For such purpose flat steel plates (Fig. 6) 2.5 mm. (0.1 in.) thick and, say, 30 by 30 cm. (11.8 in. on a side) may be used. These plates should be so held that when loaded they are stressed in bending. It was found by actual test that such plates behaved as if they were practically perfectly elastic. On these springy plates are located three plates *b* made of cork brick, the purpose of which will be indicated below, and on these is placed the vibration generator which produces forces oscillating in a vertical direction. Such a generator consists mainly of two disks *c* with eccentrically distributed masses which rotate about parallel shafts held in a rigid frame. In such an installation the horizontal components of the centrifugal forces are eliminated and only pure vertical vibrations remain. The frame of the vibration generator carries the base plate *d* which transmits the vibrations to the cork-brick plates; it is also provided with stiffening girders supporting weights *e*. By changing the size of these weights and the number and thickness of the cork-brick plates *b*, it becomes possible to do away with undesirable vibrations, for example, lateral vibrations of the whole installation. This also provides a means of changing the magnitude of the forces acting on the foundation. The machine is driven by the electric motor *f*. The compression of the spring plate is measured by an optical indicator by means of a mirror, 1, the rotation of which (Figs. 7 and 8) is proportional to the change in shape of the springy plate and hence the forces acting thereon. The ray of light which arrives in a horizontal direction is thrown upward by the mirror 1 on to a second mirror 2, which indicates the motion of the foundation with respect to a sufficiently large stationary mass and also projects the ray of light in a horizontal direction on to a sheet of sensitized paper where it is photographed as a point of light. The axes of the two mirrors cross each other vertically and their masses are so small that they can keep pace with very rapid motions. When oscillations arise in the foundation the first mirror causes the rapidly moving point of light to appear as a horizontal straight line and the second mirror as vertical straight line. If both mirrors oscillate, then the resultant of the motions is an inclined line, provided the mirrors are in phase. If mirror 2 lags in phase as compared with mirror 1, then instead of an inclined line an ellipse is produced, the shape of which makes it possible to determine the lag in phase and the magnitude of the beat of the mirror and hence the force acting on the foundation and its motion. The area of the ellipse is a measure of the energy consumed in each vibration.

The stationary mass with respect to which the motion of the foundation is measured is the weight *g* (Fig. 5) held on a base *i* located on the foundation by means of a spring stirrup *h*. The mirror measuring the motion of the foundation is located between the base *i* and the weight *g*. At frequencies from 800 to 2200 per min. the small motions of the mass *g* may be neglected.

By considering the ellipses of vibration at various frequencies one obtains from them point by point the foundation function. Such tests have been carried out on an approximately square concrete plate about 7 m. (23 ft.) long on a side and 27 cm. (10.5 in.) thick, the points under consideration lying in the middle of the slab as well as in the middle of one of the halves. The group of ellipses of vibration obtained in these tests is shown in Fig. 9 where the figures above the ellipses denote the revolutions per minute.

Fig. 6

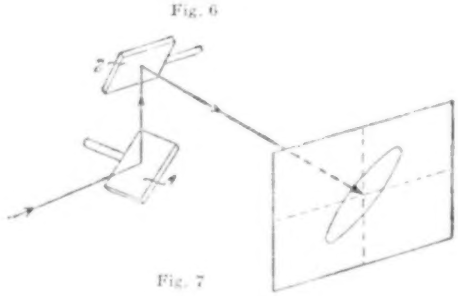


Fig. 7

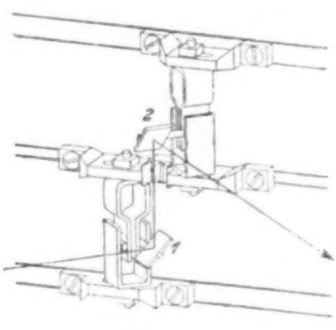


Fig. 8

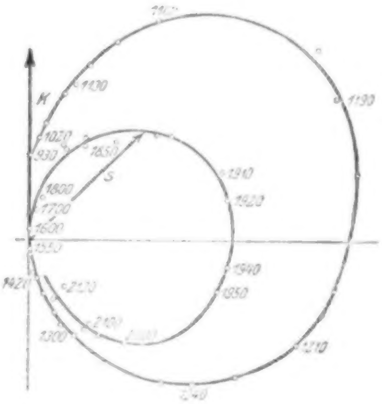


Fig. 10

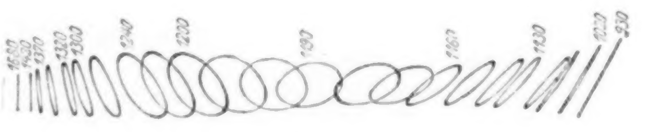


Fig. 9

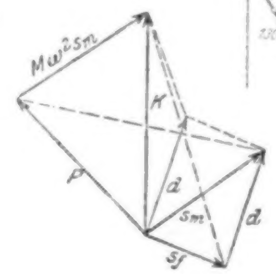


Fig. 11

FIG. 6 ELASTIC SPRING PLATE MADE OF SHEET STEEL; FIG. 7 MIRROR ARRANGEMENT FOR INVESTIGATING VIBRATIONS IN FOUNDATIONS; FIG. 8 ARRANGEMENT FOR INVESTIGATING THE BEHAVIOR OF DAMPING INTERMEDIARY LAYERS; FIG. 9 A SERIES OF OSCILLATION ELLIPSES FROM A CONCRETE PLATE; FIG. 10 ω FUNCTION FOR A CONCRETE PLATE; FIG. 11 GRAPHICAL COMPUTATION OF VIBRATIONS IN FOUNDATIONS

The foundation function for the middle of one of the halves of the plate at various speeds is given in Fig. 10. As shown there, it consists of two consecutive slices corresponding to the fundamental oscillation and the first upward oscillation of the plate, where the plate has a junction line in the middle. The component of the amplitude normal to the direction of force *K* represents the measure of the transmitted energy of vibration. The arrangement described is suitable only for the most important vibrations propagating in the vertical direction, but it may be easily applied to the other freedoms of motion.

Effect of an Elastic Intermediary Layer Between the Machine and the Foundation. Such an intermediary layer when acted on by sinusoidal forces undergoes sinusoidal changes of shape. If it is perfectly elastic the compression is in phase with the force, but if a part of the work of deformation is converted into heat the deformation must lag behind the force. As a rule such an intermediary layer has both springy and damping properties.

In order to investigate the behavior of intermediary elastic layers an arrangement was employed similar to that described above for testing the behavior of foundations. In the course of these tests a perfectly elastic steel plate is laid on a sufficiently rigid foundation layer and upon this a plate of the same size made of the material under investigation, for example, rubber, cork, etc., the whole being submitted to the action of a periodic force. The two plates are connected with a double mirror arrangement such as has been described above, mirror 1 measuring the compression

of the perfectly elastic plate and hence the force, and mirror 2 the change of shape in the plate of the material under test (Fig. 8). From the path of the point of light, which, in the case of imperfectly elastic materials, is an ellipse, and with the knowledge of the amplitude of the force, it becomes possible to determine the deformation with respect to magnitude and phase.

Tests with plates 30 by 30 cm. (11.8 in.) and about 4 cm. (1.57 in.) thick have given under static load an angle of lag of phase of 13 to 18 deg. for rubber, of 6 to 7 deg. for cork brick and of 3 to 6 deg. for natural cork. To these values correspond the following figures of the percentage of conversion of the work supplied for the purpose of changing the shape of the body into heat, namely, 23 to 31, 10 to 12, and 5 to 10, respectively. The larger angles of lag are for higher static loads. Furthermore, it was found that the springy action of the materials under investigation, i.e., of the component of deformation parallel to the direction of force, has been, in the range of frequencies used of 800 to 2000 per min., only from one-half to one-third that which could have been expected, judging by the curves of elasticity obtained at slow changes of load.

Calculation of Oscillations in a Practical Case. A machine of mass M in which there is a sinusoidal vertical force of amplitude P and frequency ω is placed on a foundation having a given experimentally determined foundation function with an intermediary elastic layer of known properties between the machine and the foundation proper. In order to determine the oscillations that arise and the energy of oscillation loss, we take from the graph of the foundation function the amplitude s_f of the foundation (magnitude and direction, Fig. 11) corresponding to force $K = 1$ and frequency ω. The application of force 1 produces in the elastic intermediary layer a deformation d. If we add vectorially d to s_f, the sum s_m gives the motion of the machine which produces force 1 at the foundation. In order to produce this motion in the phase of the inertia resistance of the mass of the machine it is necessary to exercise a force $+ M \dfrac{d^2 s_m}{dt^2}$, for which, in the case of purely periodic processes, may be substituted $-M\,\omega^2 s_m$. If we add this amount to K, we obtain in magnitude and direction the force P which must be applied to the machine in order to produce in it the assumed motion. This value, of course, is not the same as that which really acts in the machine. It is only necessary, however, to magnify or reduce the diagram proportionately or change its scale in order that P shall assume the value necessary to produce the desired oscillation process. The energy given off by the machine at each oscillation is represented by the triangle Ps_m, while the triangle Ks_f represents the energy transmitted to the foundation. The difference between these two or the triangle Kd is converted into heat in the elastic intermediary layer. If the machine has more than one freedom of motion the above calculation should be made for every one of them. Because of symmetry, however, the majority of machines have only few freedoms of motion of importance.

For other frequencies the diagram may be plotted in a similar manner, provided only the corresponding values are taken from the foundation function. By plotting the same kind of a diagram it becomes easy to investigate the probable influence of an increase in the mass of the machine or the greater softness of the intermediate layer. The problem may be solved analytically by using complex numbers instead of vectors. (*Zeitschrift des Vereines deutscher Ingenieure,* Nachlieferungsstück (Appendix), vol. 67, no. 2, Jan. 13, 1923, pp. 33–35, 11 figs., *etA*)

Short Abstracts of the Month

AERONAUTICS (See Internal-Combustion Engineering)

ENGINEERING MATERIALS

A NEW ZINC-BASE DIE-CASTING ALLOY, Charles Pack. Some die castings are subject to swelling, warping, and deterioration in service. It is claimed, however, that these troubles are attributable to the alloy used in a given casting and not to the die-casting process. In fact, it is claimed that every case of deterioration of die castings known to the author, who is connected with the Doehler Die Casting Company, has been limited to die castings made from zinc-base alloys, and he gives a brief bibliography on the subject of the deterioration of these alloys. The article claims that the Doehler Die Casting Company has developed a non-corrosive permanent zinc-base die-casting alloy which they call Ni-chro-zink. The composition of the alloy is not disclosed further than by its name. Several tests of the material are described and it is claimed that castings will show a tensile strength of approximately 18,000 lb. per sq. in. and an elongation of 4 per cent, and that fine threads, small holes, and slots can be cast without any difficulty. (*The Metal Industry,* vol. 21, no. 2, Feb., 1923, pp. 53–54, 3 figs., *d*)

FUELS AND FIRING

TEMPERING COAL AND DRAFT REQUIREMENTS, Thos. A. Marsh, Mem. A.S.M.E. In an investigation made with western coal the author found that proper tempering helps combustion. He describes in some detail his methods of testing and sets forth the following conclusions which he has reached:

1 Added moisture is beneficial with many coals, certainly with most western coals.

2 Properly tempered coal has less resistance to air flow than either very wet or very dry coal.

3 Very wet coal has less resistance to air flow than very dry coal.

4 The theory of steam formation cracking open pieces of coal does not seem to be correct.

5 Properly tempered western coal burns much more rapidly than coal either too wet or too dry.

6 Properly tempered coal burns to a cleaner ash by decreasing the fuel-bed resistance, thereby reducing the burning of holes in the fuel bed.

7 Properly tempered coal causes less siftings than dry coal, therefore fewer holes occur in the fuel bed and less coal has to be rehandled.

Knowing, therefore, that the advantages of adding moisture to many coals far more than outbalance the disadvantages, and having the above brief study of the action of moisture on air flow, some thought can be given how best to add moisture or temper coal.

Only certain general observations can be made on this subject at present, these being based on the experience of many plants with a great variety of coals. Such observations are:

a Moist coal such as obtained when a car of coal has been rained on the previous day, is usually well tempered.

b Extremely wet coal does not burn as well as coal with medium moisture.

c Moisture added an hour or two before burning is more beneficial than moisture added in stoker hoppers.

d The most usual place to temper coal is at the crusher or in the conveyor. The moisture then has time to mix thoroughly with the coal and agglomerate the fine particles.

e With western coals, one of the important features of plant operation is tempering the coal. With a known coal, intelligent firemen soon become expert in judging the best amounts of moisture for proper tempering. This varies with different coals and coal sizes.

f Exhaust steam for tempering purposes gives very satisfactory results and is used quite extensively. (*Power Plant Engineering,* vol. 27, no. 4, Feb. 15, 1923, pp. 215–217, 3 figs., *ep*)

HYDRAULICS

EXPERIMENTS ON LOSS OF HEAD IN VALVES AND PIPES OF 1/2 IN. TO 12 IN. DIAMETER, Prof. Chas. Ives Corp, Mem. A.S.M.E., assisted by Roland O. Ruble. Results of 2200 tests on 48 different gate and globe valves and 425 tests to determine pipe friction.

The loss of head due to gate valves 1/2 in. to 12 in. in diameter was measured for various openings, namely, 1/8, 1/4, 3/8, 1/2, 3/4 and fully open. The loss of head due to globe valves 1/2 in. to 2 in. in diameter, was determined under fully open conditions. As a part of the valve experiments the loss of head in pipes 1/2 in. to 12 in. in diameter was determined, and these results are also included in this bulletin. A bibliography of the subject is appended.

The loss of head due to valves and other fittings occurs in part within the valve or fitting and in part as an added loss in the pipe line downstream where normal flow has been disturbed.

Measurement of the loss of head where the downstream piezometer is attached too near the valve will give a loss in excess of that actually produced. From 20 to 25 pipe diameters beyond the valve will probably give the best position for the downstream piezometer opening. It is undesirable to have a greater length of pipe in the gage length than is actually needed to include all valve loss.

The loss of head in new, clean wrought-iron pipe from $^1/_2$ to 12 in. in diameter is given approximately by the formula $H = \frac{0.0319}{d^{1.16}} v^{1.9}$ in which H is the loss of head in feet per 100 ft. of pipe, v the velocity of flow in the pipe in feet per second, and d the pipe diameter in feet.

Globe valves offer from 15 to 40 times the resistance of gate valves for the same size. This ratio increases with the increase in the size of valves.

The length of straight pipe of the valve size which will produce the same loss of head varies from $^3/_4$ to 4 ft. for fully open gate valves and from 20 to 35 ft. for fully open globe valves. In the case of globe valves the smaller valves are equivalent to the greater length of pipe measured in pipe diameters.

The influence of location of downstream piezometer on valve loss is discussed in an appendix. The conclusion reached is that a valve causes disturbed flow in the pipe line downstream for some distance from it and that a piezometer opening which is connected so as to be in this region will be affected, giving a false record of the pressure within the pipe.

Where there is considerable change of section in passing through the valve the pressure head absorbed at that point to produce the higher velocity of flow is partially recovered in the pipe downstream.

Further, if the loss of head due to flow through a valve be obtained from a piezometer within two or three diameters below the valve, the result will be too large. Not until at least fifteen or twenty diameters of straight pipe intervene between the valve and piezometer will the reading show the true loss.

It follows from this that it is desirable to place the downstream piezometer probably 20 to 25 pipe diameters downstream from the valve. (*Bulletin of the University of Wisconsin, Engineering Series*, vol. 9, no. 1, 1922, 132 pp., 55 figs., *eA*)

INTERNAL-COMBUSTION ENGINEERING (See also Railroad Engineering)

New Fuel-Injection Valves for Compressorless Diesel Engines, Herr Roick. One of the great difficulties that have to be overcome in the development of a small high-speed Diesel engine is the variable metering of small amounts of fuel and their atomization within the very short time available in the course of each cycle. This led to intensive work in the development of proper fuel-injection valves and the present article describes the design proposed by Griev and Livens in their British and German patents. (*Motor und Auto*, vol. 17, nos. 23–24, Dec. 20, 1922, pp. 285–286, 2 figs., *d*)

Statax Three-Cylinder Rotary Engine. An interesting power plant on account of its possible use in "auxiliary" sailplanes has just been produced in Germany. This is the "Statax" three-cylinder rotary two-stroke engine which has been designed and built by F. J. M. Hansen. It is stated to weigh but 17.6 lb. complete with propeller and to develop 7.5 hp.

The chief feature of the Statax is that there is no crankcase but merely a cylindrical induction chamber to which the cylinders are bolted, with their closed ends turned inward. The outer ends of the cylinders are open, and the pistons have their skirts pointing outward. It is claimed that the extremely light weight is a result of so designing the engine that all the most highly stressed members work in tension. From the very brief particulars available (from *Flugsport* of Nov. 15, 1922), it is not quite clear how the force of the explosions and centrifugal force is transmitted from the flat steel straps to the connecting rods or their equivalent. It would appear that, instead of connecting rods, the skirts of the pistons are extended outward, the two extensions carrying lugs for the steel straps coming out from the boss which represents the crankpin. These steel straps are bent over the lugs and then twisted so as to be brought edge-on to the air as the engine is turning.

There are no valves, induction and exhaust ports of usual type, closed and opened by the pistons, being used instead. The combustion chamber is at the inner end of the cylinders, and the mixture is transferred to the combustion chamber through bent induction pipes. Centrifugal force is relied upon to get the charge from the induction chamber out of the cylinders.

The cylinders are of steel, machined from forgings, and are provided with fins of the usual type. The pistons are made of aluminum alloy, as is also the induction chamber, which, owing to the tie rods by which the cylinders are held down, does not have any great centrifugal force to resist. Also, instead of the explosion pressure being added to that of centrifugal force on the cylinders, as in ordinary rotaries, it acts in opposition to centrifugal force.

The propeller blades, of which there are three, are bolted to the induction chamber in the spaces between the cylinders. It would, for this reason, appear to be impossible to cowl-in the engine and as the combustion chambers are near the foot of the cylinders, probably this portion will require ample cooling, so that a cowl could not be used in any case.

The magneto fitted is a Bosch of the smallest type, and the plugs are Bosch-Gnome-Lilliput. The magneto is mounted on the back of the engine as is also the oil pump.

The engine is started by injecting a few drops of the fuel through the exhaust ports, when, on swinging the propeller the engine usually starts after one-half turn. Owing to the centrifugal induction system it has been found that the engine will not function with any regularity at speeds below 500 r.p.m. The maximum revolutions (for long periods) is 1600 r.p.m. At that speed it may be assumed that the propeller efficiency will not be very good in a slow machine.

Following are the main dimensions, etc., of the Statax: Bore, 60 mm. (2 in.); stroke, 70 mm. ($2^3/_4$ in.); speed, 1600 r.p.m.; fuel consumption, 0.705 lb. per hp. per hr.; oil consumption, 0.07 lb. per hp. per hr.; total weight, including a propeller of 4 ft. 11 in. diameter, 17.6 lb. These figures, it is stated, are guaranteed. (*Aviation*, vol. 14, no. 9, Feb. 26, 1923, p. 244, *d*)

The Keith-Whatmough Carburetor

New System of Carburation, G. Keith and W. A. Whatmough. Description of a new type of carburetor which was at first developed as a burner for furnaces using gas as a fuel. Fig. 1 illustrates the apparatus in diagrammatic form.

The air blast is connected at A, and is usually controlled by a valve B. The gas supply is connected to the pipe S, and the mixture flows to the burners along the pipe C. The gas supply is controlled by a balanced valve V connected to a sensitive diaphragm D working in a suitable casing. This diaphragm, which is in itself quite light, is placed vertically, so that even its own weight does not influence its action, and it is loaded on one side by the pressure produced in a tube L by the flow of mixture across the end N, which projects into the mixing tube beyond the throat or constriction M. The end N is cut obliquely, and may be partially rotated in the mixing tube, so that the pressure produced in the tube L with a given flow of mixture may be adjusted to suit the limitations of the gas supply pressure. The gas passes by the valve V until the pressure on that side of the diaphragm is in equilibrium with the loading pressure on the other side. Under this condition there is produced a pressure difference across the obturator O which bears a definite relation to the static and velocity effect produced at the moment by the mixture on the end of the tube L.

It would take up too much time to explain fully the complex variations which take place, but it will suffice for the present purpose to state that so long as the loading pressure on the diaphragm is not greater than the gas pressure existing at S, the proportion of gas to air in the mixture will be constant. The particular quality desired is obtained by adjusting the obturator O, so called to distinguish it from other valves in the apparatus. The quantity of mixture is varied by opening or closing the valve B in the air supply.

This was modified to make it suitable for internal-combustion engines still using gas as a fuel. When, however, it was applied to engines burning liquid fuel it began to boil off the liquid, and, what is more, to do it in such a manner as to prevent boiling off the lighter constituents and leaving the heavier. It was decided to carry on this operation piecemeal in a form of flash boiler capable of working with the heat which would be available from the exhaust.

The first successful type of boiler was of a cascade form in which the fuel was introduced at the top and made to trickle downward over a surface of considerable area, while the exhaust was passed in the opposite direction. By means of baffle plates sufficient turbulence was set up in the gases to enable them to give up heat to the plates of the cascade arrangement so that the liquid was completely evaporated before reaching the bottom of the boiler. This arrangement gave a good deal of trouble and was eventually replaced by the one shown in Fig. 2. In this boiler the exhaust from the engine enters at *A*, and, after impinging on the casing *B*, which contains the governing mechanism to be described later, passes in the direction of the arrow, downward, to the boiler proper. This consists of a metal plate *P* carrying a number of staggered square pins *C* between which the exhaust passes, and thence upward to the exit *D*. The staggered pins extract a maximum amount of heat from the exhaust while interposing little resistance to its passage. Depending from the plate *P*, and in thermal continuity with the heated pins is a block *E* intersected by a number of vertical saw cuts. A cross-gasway *F* connects the tops of these saw cuts to an upright gasway leading to the governor. The depending portion is surrounded by a box *G*, the lower part of which is preferably filled with metal chips. The whole boiler is surrounded by an outer casing *H*, which is lagged with slag wool below the level of the plate *P*. Fuel is admitted by the pipe *J* at the bottom of the box *G* from the fuel tank, which may be either of the gravity or pressure type. Assuming that the boiler has been heated up

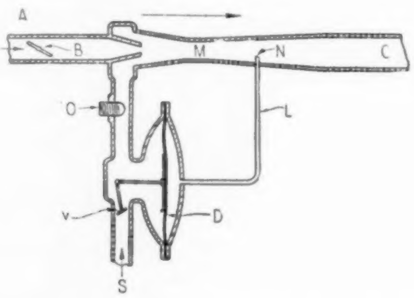


Fig. 1 Gas-Furnace Carburetor-Type Burner

sufficiently and fuel is admitted at the bottom, it will rise through the metal chips until it reaches the lower portion of the block *E*, which is brought to a point in the center. If the demand for gas is very small, the fuel just touches the lower portions of the saw-cut block and a pressure is set up equal to the head of the fuel entering at *J*, so that no more will enter. If the demand increases, the level of the liquid rises until it is in contact with a sufficiently large area of surface to give the required quantity of gas. Should the demand be suddenly reduced, the pressure of gas will rise sufficiently to depress the fuel until its area of contact is reduced sufficiently to meet the requirements.

As regards carbon deposits, it was found that none are formed in boilers where completely volatile fuel is boiled without the access of air. In reference to materials it is stated that for light and mixed fuels with equilibrium boiling points not exceeding 200 deg. cent., the boiler proper gives quite satisfactory results if made of cast iron. For fuels with an equilibrium boiling point up to, say, 225 deg. cent., such as Borneo kerosene, brass is a suitable material. With the heaviest practicable fuels, such as lamp oil, with an equilibrium boiling point of, say, 245 deg., an aluminum boiler gives the best results. The reason for the use of different materials is that for the heavier fuels a relatively better thermal conductor is necessary, while with the lighter ones a poorer conductor will suffice.

Means are provided in the boiler to prevent its flooding with the liquid when the latter is not fully vaporized; means also had to be provided to govern the gas. In this case the most satisfactory and practical device was found to consist of a heavy tube with suitable ports. The success of this device is ascribed to the unexpected formation of a kind of soft graphite on the working surfaces, which, with the continuous small movements, due to the pulsation of the engine, produces a beautifully smooth surface which moves with the minimum of friction under the high-temperature conditions prevailing.

Among other things, tests have shown that when an economical mixture was being used on heavy load it was not possible to keep the engine running satisfactorily on a light load or idling, which was found to be due to three causes, the combined effect of which was to reduce considerably the temperature of the exhaust heat, and to the fact that under low-compression pressure a weak mixture failed to ignite, although capable of being fired under high-compression pressure. This led to the development of a device

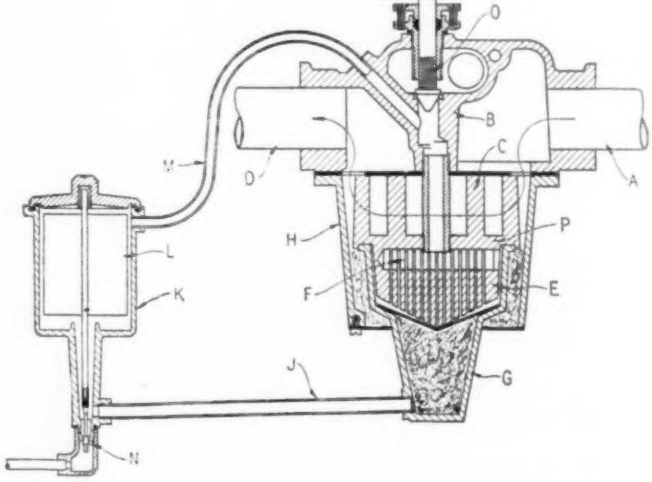


Fig. 2 Fuel Boiler with the Keith and Whatmough Carburetor

for compensation for low-compression pressure when the engine is throttled beyond the mixing point. An auxiliary carburetor is provided to start the engine and heat up the boiler.

The original article reports temperature and road tests. (Paper before the Institution of Automobile Engineers, abstracted through *The Automobile Engineer*, vol. 13, no. 173, Feb., 1923, pp. 55–59, 14 figs., *dA*)

MOTOR-CAR ENGINEERING (See also Railroad Engineering)

A Variable Gear for Motor Cars

Automatic Variable Gear, Eric W. Walford. Description of a mechanism invented by G. Constantinesco, known as the originator of the C.C. gun-control scheme that was used on practically every military aeroplane during the last two years of the war and of wave-power transmission (see Mechanical Engineering, January, 1923, p. 55).

One of the simplest forms of the mechanism is shown diagrammatically in Fig. 3. The shaft of the driving member of prime mover is shown at *A*, and that of the driven member at *B*. A crank or eccentric *C* on the shaft *A*, by means of a connecting link, causes a lever *D* pivoted at its lower end to oscillate angularly about its pivot. At its free end this lever has attached to it a pair of links *E* which at their other ends actuate pawls *F*, angularly movable around the driven shaft *B*, and adapted so that one acts on the forward swing of the lever *D*, while the other acts on the reverse swing, thereby giving a substantially continuous driving action.

So far as it is described there is nothing novel in the mechanism, nor does it act in any way as a variable gear but as a fixed-ratio transmission device only. The central feature of the invention, however, is that the lever *D* is not pivoted to a fixed, but to a movable member, here indicated by the pendulum *G* oscillating about

its suspension point at H and carrying the pivot of the lever D at some suitable distance from the point of suspension.

In its broadest and simplest conception the transmission mechanism thus employs between the driving and the driven elements a mechanical device for the storage of energy. This device has a floating connection in the drive, such that if the output of the driving member initially is insufficient to overcome the resistance of the driven member, the energy of the former is stored kinetically in the intermediate receptacle of energy, while the speed of the driving member increases automatically until this stored energy is sufficient to overcome the resistance of the driven member. Conversely, if the resistance of the driven member falls, then a portion of the energy stored between the driving and the driven member is yielded up to the driven member, the amount taken for storage from the prime mover is diminished, and the energy of the latter

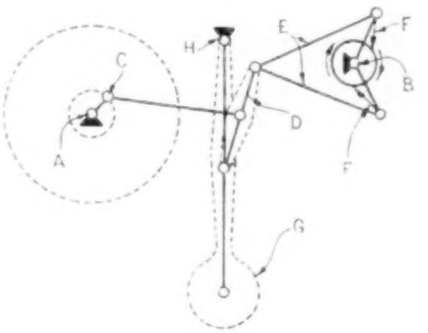


Fig. 3 Diagrammatic View of Constantinesco Automatic Variable Gear for Motor Cars

is then transmitted wholly or in part to the driven member, thus producing in it a corresponding increase of speed.

The action of the mechanism can best be understood by considering that the shaft A is set in motion and that the shaft B is at rest, and (as, for example, would be the case in a motor vehicle) would require a relatively large starting torque to set it in motion. Assuming, therefore, that the driving shaft A is not able to supply by a direct drive (such as would exist if the fulcrum of the lever D were on a fixed member) the torque necessary to start the shaft B in motion, the upper end of the lever D becomes the fulcrum for its angular movements under the operation of the crank C, and the pendulum G commences to swing. Incidentally, it may be noted that the angular motion of the pendulum under these differences is limited to a definite amount, just as would be the motion of the lever D if the pendulum were stationary.

As is well known, the characteristic of a moving body is that the energy stored in it is proportional to the square of its angular or linear velocity. Consequently, the energy supplied by the driving member, which the driven member at first is unable to utilize because it is delivered with insufficient torque, is stored up in the pendulum G. As, therefore, the driving member of prime mover, under these conditions, may be regarded as running light, its speed will increase during the storage of the energy. Here, again, it should be noted that in oscillating movements there is a natural frequency peculiar to the characteristic dimensions of the oscillating member, and that when the rate of oscillation is made otherwise than that which is natural, the frequencies are said to be forced. Such forced frequencies are now produced in the driving of the pendulum, and when its accumulated energy is sufficient, the magnitude of the reaction thrusts on the end of the lever D will cause the latter to move to and fro sufficiently to actuate the pawls F and to start the shaft B in motion.

Such movements of the pawls, at first, will be slight, corresponding to low gear, the ratio of which is accentuated by the speeding up of the prime mover during the storage of energy in the pendulum.

When motion of the shaft B has commenced, and supposing the car to be moving along the level, the resistance will diminish sufficiently for the energy of the driving member so that the supply will now be shared between the driven shaft and the pendulum. The proportion which each receives automatically varies with the resistance to be overcome, the pawls in like manner vary in their angular strokes, and the gear ratio rises or falls automatically in conformity therewith, and with the changes of speed which perforce take place in the prime mover.

The original article gives curves showing the relationship of the driving and driven members under varying conditions of speed and torque. The driving pawls are not of the ordinary tooth type but consist of pads curved around the face of a drum attached to the driven shaft. Their arrangement is such as to insure a secure grip and smooth action, which would not be readily attainable with an ordinary ratchet-and-pawl mechanism.

The most serious objection to the system described as viewed by the author appears to lie in the unbalanced forces set up by the movement of the pendulum, as it is apt to create serious vibrations. (*The Automobile Engineer*, vol. 13, no. 173, Feb., 1923, pp. 52–53, 3 figs., *d*)

POWER-PLANT ENGINEERING

Air-Cooled Condensers. Abstract of patent obtained by the Swedish Ljungström Company. The inventor states that it is sometimes difficult in connection with steam locomotives for the fans used to draw the air through the condenser. He therefore places the fans so close together that they overlap, as shown in the drawing, and the fan blades intermesh more or less after the fashion of the teeth of gear wheels. The fans must, of course, rotate in opposite directions. (British patent no. 189,718, May 1, 1922, published Dec. 7, 1922, abstracted through *The Engineer*, vol. 135, no. 3500, Jan. 26, 1923, p. 107, 1 fig., *d*)

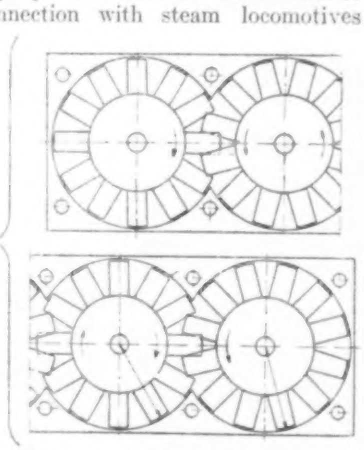


Fig 4 Ljungström Air-Cooled Condenser

A High-Pressure Boiler with Rotating Tubes

Swedish Boiler with Rotating Tubes for 1500 Lb. Pressure, Edvin Lundgren. There are places where an increase of the initial pressure of steam from 300 to 1500 lb. per sq. in. would increase the power obtained from a given weight of steam by from 20 to 60 per cent, and in the case of back-pressure engines or turbines, even 100 per cent or more. With water-tube boilers, however, at least in stationary practice, 350 lb. is considered to be generally the safe limit. In very small boilers steam was produced at higher pressures chiefly by using spiral tubes of a small diameter. There, however, the difficulty of removing sediment proves to be an important obstacle.

There is another factor which, although it has not received much attention, plays an exceedingly important part in the proper service of a boiler, particularly a high-duty boiler. This is the fact that the heat transmitted through the tube walls is far more effectively carried away by water than by steam. In boilers of the ordinary design, equipped with either straight or spiral water tubes, it is, however, practically impossible to keep the inside surface of the heated tubes steadily in contact with the water. On the contrary, the steam generated tends to linger at or near the walls in the form of bubbles, thus increasing the resistance to the flow of heat from the walls. This is, of course, more serious the higher the temperature of the fire and the higher the pressure (and hence the temperature) of the steam produced.

To meet this situation, J. V. Blomquist, a Swedish engineer, designed a new boiler called the "Atmos" in which steam is generated in rapidly revolving tubes. This rotation results in the inside circumference of the tube being entirely covered by a shell of water which is pressed against the walls by centrifugal force. This, in turn, forces the steam bubbles rapidly from the walls into the open central space from which the steam passes out through a central tube of smaller diameter. The diameter of the rotating tubes is made as large as the pressure will permit without requiring an excessive wall thickness.

Moreover, the construction is such that the rotating tubes, or the "rotors," as the inventor calls them, may expand freely without strain. In ordinary boilers it is impossible to avoid those stresses, and it is even impossible to determine them accurately.

The effective cooling of the rotor walls and the freedom from expansion strains make possible rates of evaporation as high as 60 and sometimes even 100 lb. of water per hour per square foot of heating surface. This makes for an exceedingly compact construction.

The rotors form the most important part of the new invention. Referring to Fig. 5, the hot feedwater is admitted through the vertical pipe at the left end to the cover of the stationary bearing housing or shield and enters the rotating part at *A* in the central pipe shown, which has an outside diameter of $1^1/_2$ in. and revolves in a stuffing box. To keep the packing tight against the pressure of 1500 lb. and simultaneously avoid excessive friction, is of course a serious problem which many practical steam engineers will consider apprehensively. This problem has, however, been solved by admitting oil under pressure to the central distance ring which divides the packing longitudinally into two parts.

The packing material itself consists of ordinary packing braids. The stuffing-box gland is formed like a cap with an opening on the lower side and is tightened by a single bolt located in the center of the bearing shield. This packing has proved to be entirely satisfactory during a continuous service of over twelve months and is distinguished by its low friction. The oil consumption for a 12-rotor double boiler amounts to about one pint in ten hours.

The bearing housing on this end of the rotor contains a ball bearing in which the neck of the rotor head revolves. This rotor head is a steel casting and is fastened to the rotor by means of a solid flange on the head, which is kept in place by a number of bolts. To this flange is attached the spur-gear ring which drives the rotor. The head can be readily removed for inspection and cleaning of the interior of the rotor.

The outlet head at the right-hand end of the rotor is screwed into the rotor and then welded to it. The neck of this head, and with it this end of the rotor, is supported by a roller bearing in a manner that allows free expansion and a considerable lateral movement.

As mentioned before, the steam generated is collected in the interior of the rotor and escapes through the central steam-outlet pipe (at the right in Fig. 5) which revolves in a stuffing box of the same construction as that on the water-admission tube.

The chief purpose in designing the flange *C* like the impeller of a centrifugal pump was to furnish means for measuring the thickness of the water shell and keeping it in correct proportion to the required rate of evaporation.

This is followed in the original article by a discussion of the effect produced by variation in the thickness of the water layer and the various safety devices used.

An installation along the lines described is said to have been made at the Carnegie Sugar Refining Works, Gotenburg, Sweden. Here the steam is generated at a pressure of 900 lb. per sq. in. The boiler is said to have been in continuous daily service under rather unfavorable conditions since December, 1921, and has given satisfactory results. In this boiler the tubes are rotated at 330 r.p.m. (*Power*, vol. 57, no. 7, Feb. 13, 1923, pp. 238–241, 5 figs., *dA*)

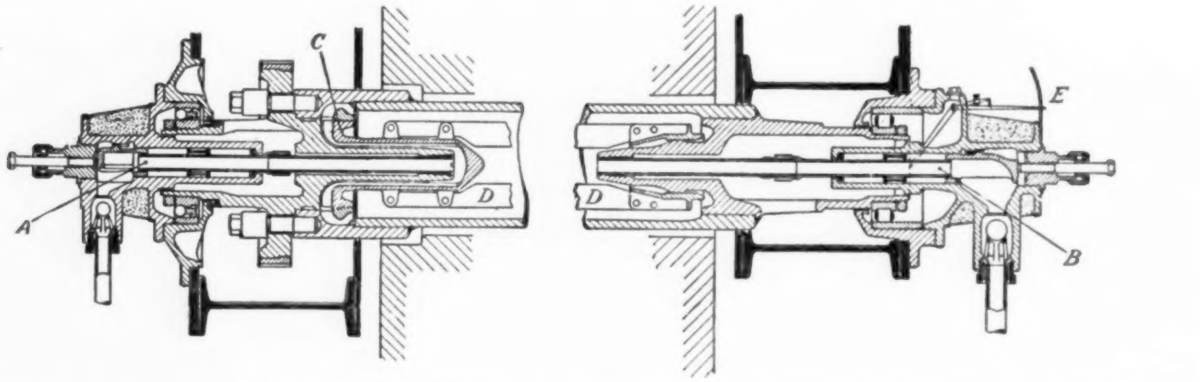


FIG 5 SECTION THROUGH A ROTOR OF THE ATMOS BOILER, SHOWING FEED CONNECTION (LEFT) AND STEAM CONNECTION (RIGHT)

POWER TRANSMISSION

SCHIEFERSTEIN POWER TRANSMISSION BY OSCILLATIONS. Data on the inventions of the engineer Schieferstein, in connection with the technical utilization of mechanical and electromagnetical oscillations. Schieferstein realizes that all oscillating systems, as for instance pistons operated by cranks, are most favorably employed for yielding useful work if their oscillations are located in proximity to their self-oscillations. In order that such engine parts may follow their self-oscillations, it will be necessary to insulate them mechanically from their driving medium. Hence a system, according to Schieferstein, consists of three essential parts: the energizer, from which the drive originates; the coupling, which transmits the drive upon the freely swinging system; and the energized or resonant part, which yields useful work.

An electrically operated quick-stroke hammer may serve as a practical example of such a system. The energizer in this case would be a crank mounted upon the end of the motor shaft. The stroke of this crank may be very short. As distinct from similar hammers which have already been constructed, the hammer piston is not connected rigidly to the crank by means of a connecting rod, but a coupling spring is interposed between the latter and the hammer piston. If now the turning speed of the electric motor and the self-oscillations of the hammer piston are tuned in respect of each other so that resonance or nearly resonance is produced between the two, the amplitudes of the oscillations of the hammer piston will be far greater than correspond to the stroke of the crank drive. As there is no power required for limiting the stroke of the hammer piston, as would be the case if it were rigidly connected to the crank, and as the reaction upon the crank drive only consists in the pressure of the coupling spring, the energy consumption of the system is said to be much smaller than if the piston were rigidly connected and the crank drive may be constructed with correspondingly smaller dimensions.

The conclusions to be drawn from the example quoted open wide prospects for the possibility of applying this system also to prime movers and power generators in which the forces of acceleration limit the increase in turning speed. The new system may also be employed to great advantage for the driving of mowing-machine knives or gate saws.

Another example shows that Schieferstein's system is applicable also to comparatively slow oscillations. This is the drive of a clock without escapement. Again he uses a crank of small stroke which may be rotated in the customary manner by the weight of the clock. The crank transmits its impulses by means of a connecting rod and through the intermediary of a coupling spring upon an ordinary clock pendulum. The crank as the energizing system and the pendulum as the resonant system must again be tuned relatively to each other in such a manner that resonance is possible. In the present case every revolution of the crank must correspond to one beat of the pendulum. If the pendulum is now set in motion it will receive through the intermediary of the spring just sufficient energy from the crank to keep up its movement, whereas, on the other hand, the coupling spring prevents the crank from causing more than one revolution during one oscillation of the pendulum. It will readily be seen that clocks

working on this system will require only a minimum of weight or spring power and will run for a long time after once being wound up. (*Engineering Progress*, vol. 4, no. 1, Jan., 1923, p. 17, *d*)

PUMPS

Condenser Extraction Pump with Glands under Pressure

Condenser Extraction Pump with Glands under Pressure. Description of a pump recently placed on the market by the Mirrlees Watson Co., Ltd. The chief feature of the new design is the elimination of all air leakage through the gland and stuffing boxes by placing these parts under a pressure corresponding to the external head against which the pump is working. A section of the pump is reproduced in Fig. 6, and serves to show the impeller arrangement and the general details of the design. Two single suction impellers are mounted on opposite sides of the pump suction *A*, and are so arranged that their inlets are facing each other, and the impellers discharge into volutes or pressure chambers *B*. These volutes are interconnected and join up to the common discharge pipe *C*. It will be noted that the two impellers work in parallel,

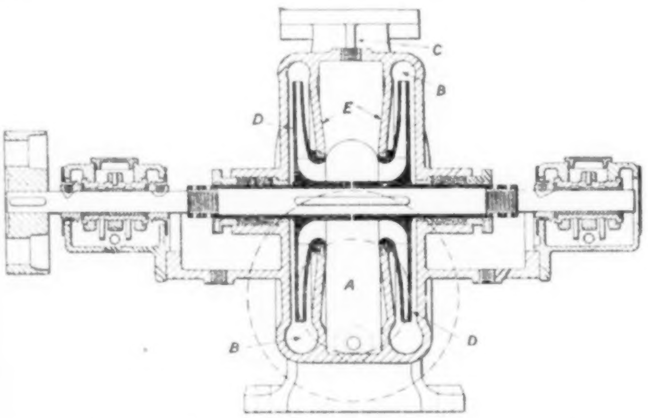


Fig. 6 Mirrlees Extraction Pump with Glands under Pressure

their disposition being such that the end thrust of the one is counterbalanced by the end thrust of the other. The chambers *D* at the opposite ends of the pump casing communicate with the inner ends of the stuffing boxes through which the impeller shaft passes, and are also connected with the volutes by means of the clearance spaces at the peripheries of the impellers. Any possible communication between the chambers *D* and the suction inlet *A* is prevented by the impellers and the diaphragms *E*. The pressure in the chambers *D* therefore corresponds to the discharge head against which the pumps are called upon to work, and as this is well above the pressure of the atmosphere there is no tendency for air to enter the pump casing through the stuffing boxes. As the design is intended for moderate discharge heads, the glands do not require to be very tight in order to prevent leakage.

The casing is split horizontally which facilitates the inspection of the working parts without disturbing the alignment of the pump and motor. The impellers are dynamically and statically balanced and the steel shaft is protected by means of a renewable bronze liner. The new pump has no water seal. The plain stuffing boxes are easily packed, and as the pressure is outwards the hot condensate is not contaminated by the infiltration of oil and grease. The type of pump described is intended for use with a surface condenser, but the design may also be applied to extraction pumps for jet condensers. (*The Engineer*, vol. 135, no. 3502, Feb. 9, 1923, p. 152, 2 figs., *d*)

RAILROAD ENGINEERING

The Next Step in Locomotive Construction, A. F. Stuebing, Mem. A.S.M.E. The author calls attention to the great improvements made in locomotive construction in recent years. He claims that as long as the reciprocating steam engine was the only type of prime mover the locomotive was also well up in the front rank of progress, and it is only since the introduction of the steam turbine and the internal-combustion engine that the efficiency of stationary power plants has shown any marked improvement over the locomotive.

Any new type of motive power if it is to be a success must be adapted to the existing railroad facilities. This means that it must haul a train load approximating that of the steam locomotive and made equal or greater speed. Any new type of motive power should be not only economical in the use of fuel, as simple and sturdy as possible, but, what is perhaps most important, it must be reliable. In a steam locomotive there are very few defects that make it entirely inoperative. It may have leaks and pounds, but it will usually bring the train in somehow.

Fuel economy and locomotive efficiency are not synonymous. The locomotive must be considered as a transportation machine. If a new type of motive power reduces the cost of fuel, but at the same time causes a greater increase in the expenditure for crew's wages, repairs, fixed charges, etc., it is not an economical transportation machine, regardless of its economy from the thermodynamic or mechanical standpoint.

From this the author proceeds to the discussion of the turbine locomotive and the Diesel locomotive, both of which apparently bear important promise for the future but neither of which the author is prepared to fully recommend for the present. (From paper entitled, Are We Due for a Radical Change in Locomotive Construction? read at the January, 1923, meeting of the New York Railroad Club, abstracted through *Railway Age*, vol. 74, no. 5, Feb. 3, 1923, pp. 323–325, *g*)

Report of the Bureau of Locomotive Inspection. Abstract from the eleventh annual report to the Interstate Commerce Commission of the chief inspector of the Bureau of Locomotive Inspection for the year ended June 30, 1922.

The report shows an increase in number of locomotives inspected, with a reduction in the number of defects and number of accidents.

During the year there were 33 boiler explosions, a substantial reduction as compared with the preceding year. Most of these explosions were caused by overheating of the crown sheet due to low water.

Of particular interest is the part of the report dealing with the applications of welding processes. Investigation of accidents where the fusion or autogenous welding process was involved is claimed to support the position previously taken that that process has not yet reached a state of perfection where it can be safely depended upon in boiler construction and repair in places where the strain to which the structure is subjected is not carried by other construction in conformity to the requirements of the law and rules, nor in firebox crown-sheet seams where overheating and failure are liable to occur, nor extensively in repairing long or numerous cracks in side sheets.

Records show that approximately 80 per cent of all autogenously welded seams involved in so-called crown-sheet failures have failed, while only 16.9 per cent of riveted seams have failed under like conditions. The fatalities where sheets tore have been $7^1/_2$ times as great as where they did not tear. From July 1, 1916, to June 30, 1922, autogenously welded seams were involved in 22.1 per cent of the crown-sheet failures, while 44.1 per cent of the total persons killed in crown-sheet accidents were killed where autogenously welded seams were involved.

A large number of accidents have been caused by defective grate-shaking apparatus and the report recommends that a power grate shaker be applied to all coal-burning locomotives. Among the other recommendations the following may be cited:

That power-reversing gears be applied to all locomotives and that air-operated power-reversing gears have steam connections with the operating valves conveniently located, so arranged that in case of air failure steam may be quickly used to operate the reversing gears.

That all locomotives, where there is a difference between the readings of the gage cocks and water glass of two or more inches under any condition of service, be equipped with a suitable water column, to which shall be attached three gage cocks and one water glass, with not less than 6 in., preferably 8 in., clear reading, and one water glass with not less than 6 in., preferably 8 in., clear reading on the left side or back head of the boiler. (*Railway Age*, vol. 74, no. 8, Feb. 24, 1923, pp. 473–474, 4 figs., *gA*)

IS THE STEAM LOCOMOTIVE OUT OF DATE? L. G. Coleman. This question is raised by the author who is assistant general manager of the Boston and Maine Railroad. His thesis is essentially as follows:

The steam locomotive today is probably the most uneconomical and unsatisfactory machine in industry. In recent years enormous improvements have been made which have very much increased its efficiency as far as power production is concerned, but these same improvements have so increased the cost and difficulty of maintenance that the time lost in terminals practically wipes out savings due to fuel economy obtained by modern devices and increased unit power.

The inherent weakness of the steam locomotive is its boiler, since when the boiler is out of commission the whole locomotive is out of commission. The limiting factor of long engine divisions is usually the boiler and not the machinery, and a machine that is otherwise ready for service always loses time in terminals by boiler-maintenance conditions, a loss which can be conservatively put down at 50 days each year.

So long as we continue to carry a portable steam boiler on each of our power plants there will be delays, such as washouts, hydrostatic tests, repairs due to leakage and numerous others.

The author believes that this situation can be made better by the use of Diesel engines than by electrification and gives the following calculation as to the former. A modern Santa Fe type locomotive with fully loaded tender weighs approximately 283 tons and will develop, say, 1800 kw. To produce the same useful output by Diesel-electric drive requires a brake horsepower of about 2600, which at 60 lb. per hp. means one or more Diesel engines of an aggregate weight of 78 tons, one or more generators not over 12 tons, a chassis to carry this load, say, 40 tons, or a total weight of 130 tons, to which, say, 20 tons should be added for radiation and accessories.

The possibilities in design are so varied as to offer many opportunities for economies, such as the use of three or four Diesel-engine generator sets mounted on a single chassis which would permit a certain freedom of electrical combinations as to voltage and amperage.

In the discussion which followed the present-day locomotive was vigorously defended, and, in particular, the great value of improvements was pointed out. Thus, it was stated that superheaters are reported to have saved their first cost in from two to three months, feedwater heaters in from 18 months to two years, brick arches in a month, and that boosters save their cost in increased tonnage revenue in from 15 days to three months.

Among other things advocated by the author is the formation by the railroads of the country of a joint bureau for research. (*New England Railroad Club*, Jan. 9, 1923, original paper pp. 190–199 and discussion pp. 199–239, illustrations in the discussion, *g*)

DIESEL-ELECTRIC RAILROAD CAR, Prof. P. Ostertag. Description of a railroad car intended to operate light trains and built by the Sulzer Brothers at Winterthur.

The car is equipped with two trucks, one of two and the other of three axles. The three-axle truck carries the Diesel engine and the electric generator, which latter is connected to the motor by a flexible coupling. The two-axle truck carries two electric motors acting on the driving shafts. They are located in a cast-iron casing and operate, through gear wheels, layshafts connected by cranks to the driving shafts.

The machinery of the present installation is much simpler than that used before. The crankshaft of the six-cylinder motor is in the longitudinal axis of the car, while the camshaft is parallel to it and runs at half the engine speed. The cylinders are in V, the exhaust pipe being located between the cylinders and the fuel tank on top of it. The Diesel engine is designed to produce 200 hp. at 440 r.p.m., but may carry a load up to 250 hp. for a short period. This gives the car a speed of 70 km. (43.5 miles) per hr. and 60 km. (37.28 miles) with a 30-ton trailer.

The operation of the engine has been simplified and air is no longer used for fuel injection, the atomization of the fuel being effected by means of a part of the explosion gases themselves. As the fuel vaporizes rapidly it ignites spontaneously without the help of either a magneto or hot tube. The compressor of the ordinary Diesel engine is therefore eliminated.

Even starting is effected without compressed air, the generator being fed from a set of storage batteries and acting then as a motor. The cooling water of the cylinders operates in a closed cycle, the hot water being carried by a centrifugal pump into a tubular radiator located on the roof of the car. Means are also provided for varying the surface available for cooling in accordance with the air temperature; in winter the hot water is used for heating the car. A thermometer placed in view of the engineer permits him to control the temperature of water at all times.

As regards the electrical part of the equipment, it is stated that the current is generated by an eight-pole machine with separate excitation and having a continuous output of 140 kw. at 300 volts.

The current to the traction motors is supplied through a Ward-Leonard distribution. The Diesel engines operate at constant speed and full load irrespective of the speed of the car, which insures the minimum consumption of fuel. The traction motors are of the series type with six main poles and six interpoles.

Numerous devices described in the original article are provided for insuring safety of operation. For example, the engineer cannot release the controller, as the circuit opens the instant he does so. These devices in the main do not differ, however, from similar apparatus used on high-speed electric traction lines in America. Numerous tests in Switzerland have shown the simplicity of operation and reliability of the new equipment. Among other things it was found that on down-grade stretches Diesel engines could be shut off, giving an additional economy in the consumption of fuel. The fuel costs generally were found to be very low as compared with similar steam equipment. It is claimed that the cost of operation of the Diesel-driven railroad car is much lower than that of a gasoline-driven car and that in addition a heavy-oil equipment is safer from the fire hazard point of view. (*Bulletin Technique de la Suisse Romande*, vol. 49, no. 2, Jan. 20, 1923, pp. 21–26, 9 figs., *d*)

SPECIAL MACHINERY (See Power Transmission)

TESTING AND MEASUREMENT

VAN WEST PORTABLE SET FOR TESTING GEARING. Description of an apparatus developed in Holland which can be used with the gears in place, provided it is found convenient to measure the axial pitch of the helical teeth rather than the circumferential.

The apparatus consists of a round bar, denoted by *A* in Figs. 7 and 8, and a square steel bar. The square bar is fixed in a groove cut parallel to the axis, and serves as a bed over which the measuring devices can be moved. With the pinion and wheel in true alignment, this square bar will always be parallel with the axes of the wheels when the bar *A* is placed as indicated in Fig. 8. The bar *A* is magnetized, and thus fits itself in between the wheels, and is held firmly without risk of distortion. A rod *C*, which can be clamped by a set screw, as best seen in Fig. 7, serves to adjust the angular position of the magnetized bar by abutting against the pinion shaft as indicated in Fig. 8. On the square bar a block, *B*, is arranged to slide. This block carried the two fingers, *E* and *D*, clearly shown in Fig. 7. The finger *E* can be clamped at any desired point of the rod *F* by the milled head shown, while the finger *D* has a tail piece which abuts against a dial micrometer as indicated. Both fingers can be rotated around or with their supporting rod, *F*, which is secured by a clamp, as shown in Fig. 7. An adjustment is provided at *H* for bringing to zero the indicator of the dial micrometer *G*. If the whole apparatus is pressed in an axial direction, the ends of both fingers come into contact with the flanks of consecutive teeth and the reading of the dial micrometer is recorded. A second set of readings is taken after shifting the apparatus one pitch forward, the operation being repeated till a complete record is obtained. Mr. Van West states that in the case of a pair of smoothly running marine gears, on which he made measurements as above described, the greatest difference found in the axial pitch was only 0.03 mm., or about 0.0012 in.

The device can also be used for testing the teeth of pinions before being put in place. To this end the pinion is laid on a flat surface, as indicated in Fig. 10, and the bar *A* held between the pinion and an angle plate, the measurements being made as before. For testing the alignment of gears the apparatus is placed as indicated in Fig.

9, and the bar turned round until one end of the edge of the square bar touches the teeth. A feeler gage, inserted under the other end, measures the error of alignment. (*Engineering*, vol. 115, no. 2977, Jan. 19, 1923, pp. 74, *d*)

Radiography of Metals at Watertown Arsenal, H. H. Lester. Of interest because of the pioneering nature of the work and unusual completeness of the apparatus available.

The radiographic process is essentially a photographic process. The metal to be radiographed is exposed between the target of a Coolidge X-ray bulb and the photographic film. After exposure the negative is developed in the ordinary way. The photograph obtained is unlike an ordinary photograph in that the latter is superior to the radiograph in the greater amount of surface detail. The radiograph, however, in addition to some surface detail shows in one picture details of both surfaces which can be secured with the camera only by taking two pictures from two sides; further, the radiograph shows details of internal structure which the ordinary photograph does not show at all. In a radiograph white spots represent thin metal. This may be due to the presence of cavities or geometry of the casting. Unfortunately, the majority of the illustrations in the original article cannot be reproduced, thus making it impossible to abstract certain parts of considerable interest.

In radiography it is important to know through what thickness of metal pictures may be taken. Thirty minutes is the upper limit of practical exposure time in the process used at the Watertown Arsenal, although in cases of special importance longer time could be given without, however, corresponding return in penetrability.

Properly developed radiographs indicate not only the presence of defects such as gas cavities but even their exact dimensions. Of still greater importance perhaps than the exact dimensions of the flaws is the question of their detectability. Experiments carried out at the Laboratory of the Watertown Arsenal indicate that where the linear dimension of the flaw, i.e., the dimension parallel to the axis of the X-ray beam, is approximately equal to $1^1/_2$ per cent of the total thickness of the metal in the region adjacent to the flaw, the image may be distinguished. For practical working conditions detectability is placed at 2 per cent, which means that a cavity 0.05 in. in diameter can be detected in metal 2.5 in. thick.

To locate a cavity within the metal the stereoscope is used. In making a stereoscopic examination the area to be radiographed is arranged so that the center of the area, the center of the film, and the target are in line; then the bulb is moved 3 cm. (1.18 in.) to one side of the center line and the exposure made. A second negative is made with the bulb 3 cm. (1.18 in.) to the other side of the center line. These two negatives after development are placed in a stereoscope so arranged that the films may be as far from the eye as they were from the target of the bulb when the exposure was made. When the machine is properly adjusted the picture stands out in bold relief. The surface markings and structural details serve to outline the picture, and the faults appear in their proper places within the material. The picture is quite striking and gives the illusion of looking at a transparent model of the casting. By reversing the negative this model may be turned completely over and one may look through the casting from the other side. The method permits locating details with reference to other details of internal structure. (*Army Ordnance*, vol. 3, no. 16, pp. 210-215, 10 figs., *deA*)

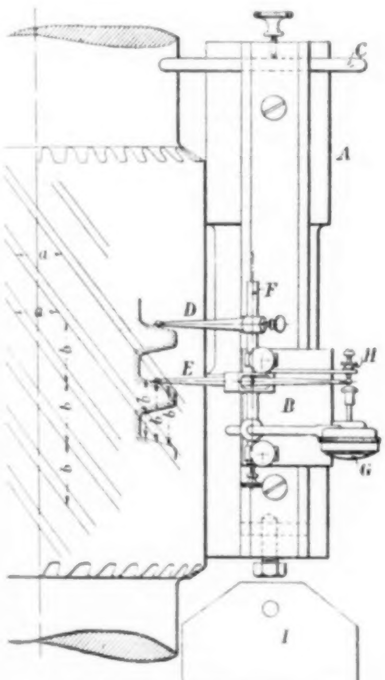


Fig. 7

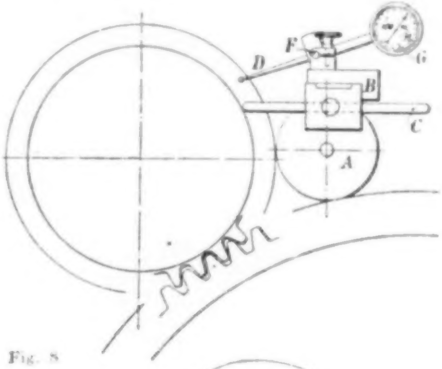


Fig. 8

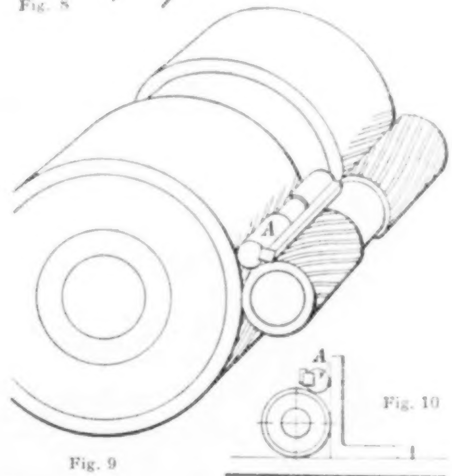


Fig. 9

Fig. 10

Figs. 7 to 10 The Van West Gear-Tooth Testing Appliance

THERMODYNAMICS

Variation of the Specific Heat of Air with Temperature. An article based on a paper read in January before the Faraday Society in London by W. G. Shilling of the East London College, based on experiments carried out by himself and Prof. J. R. Partington.

In the present experiments, the method followed was that of determining the ratio C_p/C_v from the velocity of sound in the gas at definite temperatures. The article describes the method used and gives the results obtained, reproduced in Table 1, where T is the temperature in deg. cent. absolute and V the velocity of sound in meters per sec.

The investigators do not rely upon their values at temperatures above 800 deg. cent., because with the heating tube used it proved difficult to secure a sufficiently long central tube portion of uniform temperature, and experiments are being continued with longer tubes.

For the range up to 700 deg., however, the equation $C_v = 4849 + 0.00036\ T$ (in gram calories) would hold, and that formula, which is in good agreement with previous work, together with the concordance of the observations, places the knowledge of the specific heat of air on a much firmer basis than before.

TABLE 1 VARIATION OF SPECIFIC HEAT OF AIR WITH TEMPERATURE FROM EXPERIMENTS BY J. R. PARTINGTON AND W. G. SHILLING

T	V	C_p/C_v	C_v	C_p
288	340.65	1.403	4.952	6.945
373	387.16	1.399	4.985	6.974
473	435.44	1.396	5.017	7.004
573	478.80	1.393	5.053	7.039
673	518.30	1.389	5.098	7.084
773	555.10	1.387	5.130	7.115
873	589.35	1.385	5.157	7.142
973	621.53	1.382	5.197	7.182
1,073	651.87	1.379	5.237	7.222

Professor Partington considers the figures reliable within 0.5 per cent up to 400 deg. cent. and within 1 per cent or 2 per cent for the higher temperatures. Beyond this, however, the reliability of the equation of state comes in, and possibly preference might be given to the Dieterici equation as compared with that of Berthelot, the former covering a wider range of temperatures. (*Engineering*, vol. 115, no. 2977, Jan. 19, 1923, pp. 82-83, 2 figs., *eA*)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general, *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

ENGINEERING RESEARCH

A Department Conducted by the Research Committee of the A.S.M.E.

Apparatus and Supplies for Research

IS AN INSTRUMENT for my present need available or must I design one? Where may a specified type of apparatus be obtained? Where may plans for laboratory construction and ideas about special equipment be had? What are the best sources of certain chemical or other supplies for research? These are samples of recurrent questions from the research laboratory.

With the purpose of helping to meet the growing demand for information the Research Information Service of the National Research Council has assembled catalogs and lists of research appliances issued by makers and dealers. Publications of nearly 500 domestic firms and nearly 200 foreign firms are now on file. They have been classified for effective use and a subject catalog of apparatus has been prepared. The Service has also an up-to-date list of manufacturers and distributors of research chemicals.

Mechanical engineers and investigators who desire assistance in locating special instruments, apparatus, or supplies for use in their laboratories are invited to avail themselves of the resources of this organization.

The address for inquiries is Information Service, National Research Council, Washington, D. C.

Résumé of the Month

A—Research Results

The purpose of this section of Engineering Research is to give the origin of research information which has been completed, to give a résumé of research results with formulas or curves where such may be readily given, and to report results of non-extensive researches which in the opinion of the investigators do not warrant a paper.

Aeronautics A1-23. Preparation of Light Aluminum-Copper Casting Alloys. See *Non-Ferrous Metals A2-23.*

Corrosion A1-23. Protective Metallic Coatings for the Rustproofing of Iron and Steel. This subject is thoroughly covered by Circular No. 80 of the Bureau of Standards, the second edition of which has just been issued. The nature of metallic corrosion and the principles underlying methods of prevention are discussed in the Introduction. Then follow chapters on types of coatings and methods of application, microstructure with numerous micrographs, preparation of surface before coating and accompanying effects upon the mechanical properties of the steel, methods of testing coatings, and finally recommendations concerning different methods. A selected bibliography on corrosion and its preventive metallic coatings is given as an appendix.

Apply to the Superintendent of Documents, Government Printing Office, Washington, D. C. Price 20 cents.

Electricity Utilization A1-23. Electric Brass-Furnace Practice. See *Non-Ferrous Metals A1-23.*

Forest Products A1-23. Moisture-Resistant Coatings for Wood. Technical Note No. 181 recently issued by the Forest Products Laboratory, Madison, Wis., describes an interesting investigation of the relative merits of the various methods now employed for coating wood to protect it from moisture. In a table at the end of this report 17 methods of coating wood are rated according to their efficiency.

Foundry Equipment, Materials and Methods A1-23. Electric Brass-Furnace Practice. See *Non-Ferrous Metals A1-23.*

Framed Structures. A1-23. Compression Tests of Structural-Steel Angles. This article presents the results of compression tests of 170 structural angles, made at the Pittsburgh branch of the Bureau of Standards. The object of the tests was to determine the ultimate compressive strength of angles fastened at the ends in such ways as would closely correspond to their connections in the construction of transmission towers. There was also tested a series of angles with square ends. An end fixation factor was found to represent satisfactorily the effect of different types of end connections.

This paper was prepared by A. H. Stang and L. R. Strickenberg. It is known as Technologic Paper No. 218 and may be obtained for 10 cents from the Superintendent of Documents, Government Printing Office, Washington, D. C.

Fuels A2-23. Coal Analyses from Twenty-five Laboratories Compared. The Bureau of Mines has recently conducted a study of results obtained in analyzing similar samples of coal and coke by twenty-five laboratories throughout the country, in comparison with results obtained in the Bureau's coal laboratory at Pittsburgh. The comparison is based on the average of 24 analyses by the Bureau of Mines on each kind of coal, and curves are shown giving the deviation from this average analysis, together with deviations from the check limits of the American Society for Testing Materials.

This report of Investigation, Serial No. 2432, was prepared by A. C. Fieldner, H. M. Cooper, and F. D. Osgood.

Iron and Steel A1-23. Protective Metallic Coatings for the Rust-proofing of Iron and Steel. See *Corrosion A1-23.*

Iron and Steel A2-23. Effects of Carbon and Manganese on the Mechanical Properties of Pure Iron. This paper which was prepared by R. P. Neville and J. R. Cain of the Bureau of Standards describes the preparation and mechanical properties of an extensive series of pure alloys of electrolytic iron, carbon, and manganese whose compositions were so chosen as to bring out the specific effects on pure iron of additions of manganese, carbon, and carbon and manganese together in varying proportions. The maximum content each of carbon and manganese in each series is about 1.6 per cent; the minimum, 0 per cent, or pure iron.

Address Superintendent of Documents, Government Office, Washington, D. C. Price 10 cents.

Iron and Steel A3-23. Structure of Martensitic Carbon Steels and Changes in Microstructure which Occur upon Tempering. H. S. Rawdon and S. Epstein have recently published as Bureau of Standards Scientific Paper No. 452 the results of their investigation in this field. This study of the changes in structure resulting upon tempering was made in a series of 6 carbon steels ranging from 0.07 to 1.12 per cent carbon, quenched from temperatures varying from 750 to 1250 deg. cent. and tempered for different lengths of time at 100 to 650 deg. cent. Upon quenching, martensite is formed throughout each austenite crystal in a manner strictly analogous to the freezing of solid-solution alloys. A redistribution of carbon takes place and the conspicuous martensite plates are found to be distinctly lower in carbon than the "filling material" between the plates. The enrichment of the carbon in the "filling material" may be great enough in some steels to allow small patches of austenite to persist after quenching.

This report includes reproductions of a large number of micrographs. Address the Superintendent of Documents, Government Printing Office, Washington, D. C.

Metallurgy and Metallography A1-23. Structure of Martensitic Carbon Steels and Changes in Microstructure which Occur upon Tempering. See *Iron and Steel A3-23.*

Non-Ferrous Metals A1-23. Electric Brass-Furnace Practice. In 1911 no electric furnace suitable for melting brass was in existence. The Bureau of Mines then began to study the problem experimentally and to encourage brass melters, electric-furnace designers, and electric generating stations to study it also.

The present report, Bulletin No. 202 prepared by H. W. Gillett and E. L. Mack, is published to record the progress so far made in melting brass electrically; to aid the plants which have not yet taken up such melting by pointing out the types of furnaces available, describing their performance and indicating their possibilities and their limitations; and to encourage further experimentations with and the development and installation of electric brass furnaces. It also summarizes the theoretical advantages of electric brass melting.

Eighty different types or different makes of the same type of electric furnace were used in the investigation and are described in this Bulletin of 350 pages.

Address the Superintendent of Documents, Government Printing Office, Washington, D. C. Price per copy 50 cents.

B—Research in Progress

The purpose of this section of Engineering Research is to bring together those who are working on the same problem for coöperation or conference, to prevent unnecessary duplication of work, and to inform the profession of the investigators who are engaged upon research problems. The addresses of these investigators are given for the purpose of correspondence.

Air B1-23. Flow of Air Through Orifices. See *Fluid Flow B1-23.*

Fluid Flow B1-23. Flow of Air Through Orifices. Under the direction of Associate Professor J. A. Polson of the department of steam engineering of the University of Illinois an investigation on the above subject is now under way. Special apparatus has been designed and built for the tests by F. W. Martin and F. C. Linn.

The air is being weighed in the air weighing plant and passed through the orifice. Various methods of determining the flow at the orifice are to be employed and the results compared with the weight determined from the weighing plant. The calculation of values for the coefficients of flow of the several types of orifices and short tubes employed will then be a simple matter.

Friction and Allied Subjects B1-23. Friction Pressure Drop in Ammonia Pipe and Fittings. See *Refrigeration B1-23.*

Refrigeration B1-23. Friction Pressure Drop in Ammonia Pipe and Fittings. As part of program of research in refrigeration the Engineering Experiment Station of the University of Illinois is undertaking an investigation on this subject. These experiments are being directed by Prof. H. J. Macintire and Mr. J. P. Mullen.

F—Bibliographies

The purpose of this section of Engineering Research is to inform the profession of bibliographies which have been prepared. In general this work is done at the expense of the Society. Extensive bibliographies require the approval of the Research Committee. All bibliographies are loaned for a period of one month only. Additional copies are available, however, for periods of two weeks to members of the A.S.M.E. These bibliographies are on file at the headquarters of the Society.

Ceramics and Glass F1-23. This bibliography covers the technical journal articles and Government reports which the Bureau of Mines have consulted during their investigations on this subject. It covers six typewritten pages and is known as Bureau of Mines Reports of Investigations, Serial No. 2437.

CORRESPONDENCE

Contributions to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities and policies of the Society in Research and Standardization.

Mathematical Determination of the Modulus of Elasticity

To the Editor:

I read with interest the communication from Mr. David Guelbaum on the above subject in the December issue of Mechanical Engineering, also Mr. Wm. R. Bryans' comments thereon in the February issue. Perhaps the following suggestions would be of interest.

1 Regarding limitations of theory, for a thin strip of material such as must be used in these experiments the error due to assuming an infinite radius of curvature is by far the least of the errors involved. The common theory is only 7 per cent in error when applied to hooks, where the sections are deep and the radii of the order of two or three inches. (See Elasticity and Resistance of the Materials of Engineering, by Wm. H. Burr.)

2 It must be noted that much depends on the accuracy of determination of the moment of inertia I. Suppose we use a strip of metal 0.0500 in. thick by 0.200 in. wide. Let us vary both dimensions by 0.0010 in.

$$I = \frac{b\,h^3}{12} \therefore dI = \frac{\partial I}{\partial b}\,db + \frac{\partial I}{\partial h}\,dh = \frac{h^3}{12}\,db + \frac{h^2 b}{4}dh$$

$$\frac{dI}{I} = \frac{db}{b} + \frac{3}{h}\,dh = \frac{0.001}{0.200} + \frac{3}{0.05} \times 0.001 = 0.065 \text{ or } \pm 6.5 \text{ per cent.}$$

Using ordinary machine methods, we may then be as much as $6^1/_2$ per cent in error, even though our other sources of error are negligible. For accurate work our test piece must be ground to size, within ± 0.0001 in.

3 If the type of testing machine should prove to have merits for special work, as it probably will, the form of machine described is handicapped by a complicated mathematical formula. This handicap may be removed by redesigning the machine.

We will agree that $M = EI/\rho$. If we can make $\rho =$ constant, our mathematical work becomes absurdly simple. To do this, it is only necessary to apply a *couple* at the end of the strip. The moment at all points will be constant, E and I are assumed constant, so the neutral axis becomes a circular arc.

The mechanism would be as follows: One end of the strip would be rigidly clamped in a fixed vise and the other end clamped to a vise on a movable carriage on wheels. A capstan drum on the movable carriage would have a cord about it, the ends passing over pulleys fixed to the bedplate, and supporting weight pans. A slight initial deflection would be produced by adding weights to one pan. The position of the end of the strip could be read with a microscope. More weights would be added to the pan, and the new position of the strip end determined. We may be sure of having eliminated initial position errors if $M\rho =$ constant $= EI$ (for elastic materials, of course).

Garden City, L. I., N. Y. Edward Adams Richardson.

To the Editor:

Referring to the letter in the December issue of Mechanical Engineering from David Guelbaum on the mathematical determination of the modulus of elasticity and the comment thereon by Wm. R. Bryans in the February issue, I would call attention to some of the difficulties encountered in calculating the modulus of elasticity from experiments with thin strips.

1 In bending a rod of rectangular cross-section, for example, Fig. 1 (a), the cross-section is assumed to remain plane. Now, let us denote by $d\theta$ the small angle between two neighboring cross-sections, by ρ the radius of curvature, and by δ the extension of any fiber at a distance z from the neutral axis. Then the unit elongation of the fiber under consideration is—

$$e = \frac{\delta}{\rho d\theta} = \frac{z}{\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

The corresponding stress is—

$$p = eE = \frac{zE}{\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

and integrating over the entire cross-section we obtain the well-known equation of the elastic line—

$$M = \frac{EI}{\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

Contrary to Mr. Bryans' statement, there is nothing in this equation that requires the radius of curvature to be practically infinite. The underlying assumptions are (1) that the cross-sections remain plane and (2) that Hooke's law applies. Let us see what this means. In order to be within Hooke's law the maximum stress must not exceed the elastic limit of the material. This stress we obtain from [2] for $z = +h/2$, whence—

$$p_{\max} = \frac{hE}{2\rho} \text{ or } \frac{\rho}{h} = \frac{E}{2p_{\max}}$$

Now for iron $E = 3 \times 10^7$ lb. per sq. in. and $p_{\max} = 3 \times 10^4$ lb. per sq. in., so that—

$$\frac{\rho}{h} = \frac{3 \times 10^7}{2 \times 3 \times 10^4} = 500$$

Therefore the sole proviso in our experiments is that for material such as iron the radius of curvature must be large as compared with the height of the cross-section. The curvature itself may not be large, but in that case the height of the cross-section must be exceedingly small. It follows that because of convenience we shall be obliged to experiment with thin strips of metal where the distortion of the cross-sections must be taken in consideration.

As is well known, an elongation e in one direction causes a contraction in perpendicular directions equal to e/m, where m is Poisson's ratio. Thus [see Fig. 1 (b)] on the convex side with respect to the elastic line the cross-section will be subject to lateral

contraction and on the concave side, where the longitudinal fibers are under compression, there will be lateral dilatation in the plane of the cross-section. As a consequence the cross-section distorts as shown in Fig. 1 (*b*); the neutral axis cannot therefore remain straight and it is easy to see that its curvature will be $1/m\rho$. Until the width b of the cross-section is of the same order as the height h and h/ρ is a small quantity, the additional deflection due to this curvature will be a small quantity of second order, which can be neglected. In this way the fact of the bending of the neutral axis will not interfere with the assumption that the cross-sections remain plane.

2 Now consider a thin, wide strip bent as shown in Fig. 2 (*a*). The full lines of Fig. 2 (*b*) are the boundaries of a cross-section before bending, and if we apply the foregoing considerations we shall arrive at the conclusion that the cross-section distorts as shown by the dotted lines.

Experiments, however, show that this is not the case. It will

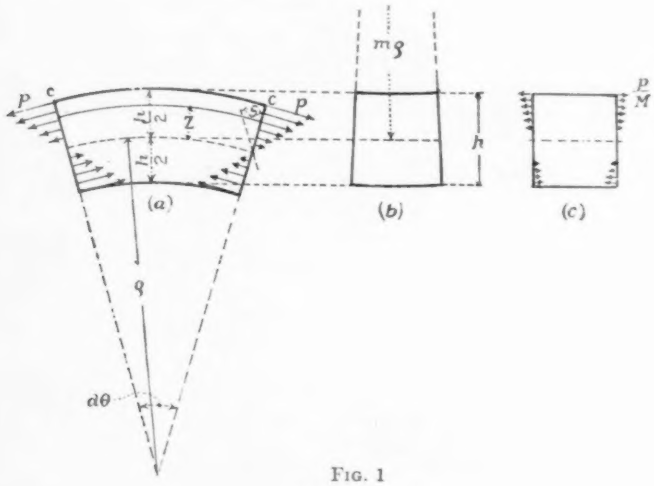


FIG. 1

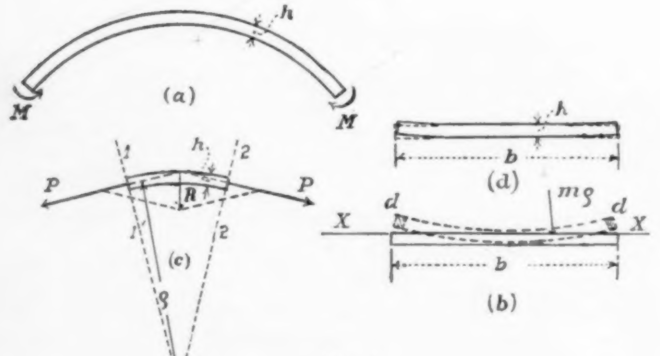


FIG. 2

be found that the middle part of the cross-section remains as before and distortion occurs practically only near the edges as shown in Fig. 2 (*d*). The reason for this behavior is plain. We must not forget our fundamental assumption, which is that bending consists of a rotation of the bar elements around the neutral axis and that the elements of a plane cross-section before bending remain in a plane after bending. This hypothesis has been proved to be substantially true for straight bars of constant cross-section. This being the case, it is evident that the neutral axis must remain substantially such a straight line as, for instance, x-x, Fig. 2 (*b*). Now it will be seen that if the distortion of the cross-section is as assumed in Fig. 2 (*b*), the parts d, d, will lie entirely in the tension zone. If we then cut a longitudinal strip from the bar near its edges, as represented in Fig. 2 (*c*), the cross-sections 1-1 and 2-2 will be subject to tensions only, whose resultants are the forces P. Their resultant is the force R, which evidently prevents the cross-section through the strip from curving upward as assumed in Fig. 2 (*b*). This at once explains why the distortion of the cross-section is more like that shown in Fig. 2 (*d*) and also confirms the statement that the fundamental hypothesis of bending is substantially true.

Since then in the bending of a thin, wide strip the cross-sections (with a negligible distortion at the corners) remain substantially as before, whereas our theory requires a pronounced distortion, it is evident that Equations [2] and [3] should be modified.

Returning to the case of Fig. 1, it is clear that the distortion of the cross-section will be prevented by applying in its plane the forces shown in Fig. 1 (*c*). These forces in turn affect the longitudinal strain. Thus for a fiber c-c, Fig. 1 (*a*), the elongation due to the longitudinal stress p alone would be p/E. On this there will be superimposed a contraction due to the stress p/m acting in the plane of the cross-section, and this contraction is—

$$\frac{p}{mE} \times \frac{1}{m} = \frac{p}{m^2E}$$

The total elongation of the fiber c-c will therefore be—

$$e = \frac{p}{E} - \frac{p}{m^2E}$$

whence—

$$p = \frac{eEm^2}{m^2 - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

Using this value of p in [2], the equation for the elastic line becomes—

$$M = \frac{EI}{\rho} \times \frac{m^2}{m^2 - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

This is the formula that should be used in the case of bending thin, wide strips. If we take $m = 10/3$, then—

$$\frac{m^2}{m^2 - 1} = 1.10$$

Hence in experiments on thin strips of comparatively great width, Equation [3] will give a value for E which is about 10 per cent too high.

3 Now Equations [3] and [5] furnish solutions of the problem of bending in two extreme cases. For intermediate cases more detailed consideration of the distortion of the cross-section is necessary. To give the complete theory of these problems would be beyond the scope of the present communication.

For the sake of completeness, however, the writer wishes to state that in the case of a strip acted upon at its terminal cross-sections by equal and opposite bending moments, there is obtained in place of Equations [3] and [5]—

$$M = \frac{EI}{\rho} \times \frac{m^2 - k}{m^2 - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

in which the quantity k is a function of the radius ρ, the thickness h, and the width b of the strip. A number of values of k are given in the table below where—

$$\beta b = b\sqrt[4]{\frac{3(m^2 - 1)}{m^2\rho^2h^2}} = 1.286\,\frac{b}{\sqrt{\rho h}} \quad \ldots\ldots\ldots\ldots [7]$$

βb =	0.5	1.0	1.5	2.0	2.5	3.0	4.0	5.0	$\beta b > 5$
k =	0.999	0.995	0.975	0.877	0.818	0.725	0.534	0.420	$\frac{2}{\beta b}$

For example, for $h = 1$ mm. and $\rho = 500$ mm., we obtain, from [7], $\beta b = 3$; the value of k for $\beta b = 3$, as given in the table, is 0.725. Substituting this value in Equation [6] gives—

$$M = \frac{EI}{\rho} \times \frac{m^2 - 0.725}{m^2 - 1} = \frac{EI}{\rho}\,(1 + 0.0272)$$

that is, Equation [3] would give a radius of curvature about $2^3/_4$ per cent too large.

The table therefore enables us to estimate in what cases Equation [3] can be used without material error in calculating E from the experiments with strips.

S. TIMOSHENKO.

Philadelphia, Pa.

Work of A.S.M.E Boiler Code Committee

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of the Society, for approval, after which it is issued to the inquirer and simultaneously published in MECHANICAL ENGINEERING.

Below is given the interpretation of the Committee in Case No. 411, as formulated at the meeting of January 9, 1923, and approved by the Council. In accordance with the Committee's practice, the names of inquirers have been omitted.

CASE NO. 411

Inquiry: Par. 212c which permitted increasing the pitch of staybolts on cylindrical surfaces over that required for flat plates, had, about two years ago, been held in abeyance pending the revision of the Boiler Code, but nothing has been left in its place. In view of this, what rules should be followed pending the publication of the revised Code?

Reply: It has been proposed to revise Par. 212c as follows, and the Committee recommends to the state inspectors that this rule be followed in place of the rules now given in Par. 212c of the Code:

Par. 212 *d* For furnaces over 38 in. in outside diameter of vertical fire-tube boilers and other types of furnaces and combustion chambers not covered by special rules in this Code, which have curved sheets subject to external pressure, that is, pressure on the convex side, the staying, both circumferential and longitudinal, shall be provided for in accordance with the following formula:

$$P = \frac{CT^2}{p^2} + 250\frac{T}{R}$$

where p and the value of C are as given in Par. 199, p shall not exceed $2T$, and p^2 shall not exceed $0.008\ CTR$.

The stress per sq. in. in staybolts shall not exceed 7500 lb., based on a total stress obtained by multiplying the product of the circumferential and longitudinal pitches by $\left(P - 250\frac{T}{R}\right)$.

Second Revision of A.S.M.E. Boiler Code, 1923

A HEARING is held by the Boiler Code Committee at least once in four years, at which all interested parties may be heard, in order that such revisions may be made as are found to be desirable, as the state of the art advances. The year 1922 became the period of the second revision and the Boiler Code Committee held a Public Hearing in connection with the recent Annual Meeting of the Society in December, 1922, to which the membership of the Society and everyone interested in the steam-boiler industry was invited to attend and present their views.

For the convenience of every one interested, a printed schedule of the various proposed revisions had been published and distributed to all those who were invited to attend the Public Hearing and the opportunity was given thereat for the most careful consideration of all of the proposed revisions. As a result of the suggestions received at the Public Hearing, a number of modifications of the previously announced revisions were offered and in addition suggestions were received for still further revisions of the Code. All of these suggestions for modifications and new revisions have been carefully considered by the Boiler Code Committee and the result in modifications of revisions and additional revisions are here published.

It is the request of the Committee that these revisions be fully and freely discussed so that it may be possible for anyone to suggest changes before the rules are brought to final form and presented to the Council for approval. Discussions should be mailed to C. W. Obert, Secretary to the Boiler Code Committee, 29 West 39th St., New York, N. Y., in order that they may be considered by the Boiler Code Committee.

The revisions here published are limited to the paragraphs appearing in the 1918 Edition of the A.S.M.E. Boiler Code, and the paragraph numbers refer to the paragraphs of similar number in that edition. For the convenience of the reader in studying the revisions, all added matter appears in small capitals and all deleted matter in smaller type in brackets.

Modifications of Revisions

PAR. 9 ADD THE FOLLOWING TO THE REVISED FORM PRINTED IN THE JULY 1922 ISSUE OF MECHANICAL ENGINEERING:

SEAMLESS TUBES OR LAP-WELDED PIPE MAY BE USED FOR DRUMS OR OTHER PRESSURE PARTS OF A BOILER PROVIDED SUCH TUBES OR PIPES CONFORM TO THE SPECIFICATIONS FOR WELDED AND SEAMLESS STEEL AND WROUGHT-IRON PIPE, AND PROVIDED ALSO THAT THE OUTSIDE DIAMETER OF THE TUBES OR PIPES DOES NOT EXCEED 20 IN.

PAR. 19 CANCEL PROPOSED REVISION OF THIS PARAGRAPH PRINTED IN THE DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING.

PAR. 21 REPLACE REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING WITH THE FOLLOWING:

21 *Tubes for Water-Tube Boilers.* The maximum allowable working pressures for STEEL OR WROUGHT-IRON tubes used in water-tube boilers, shall be for the various diameters and gages measured by Birmingham wire gage, as given in Table 2. REDRAWN PIPE NOT TO EXCEED 1 1/2-IN. STANDARD PIPE SIZE WHICH MEETS THE PIPE SPECIFICATIONS, MAY BE USED FOR WATER-TUBE BOILERS FOR A WORKING PRESSURE NOT TO EXCEED 200 LB. PER SQ. IN., WHEN SCREWED IN THE SHEET, PROVIDED THE WALL THICKNESS IS AT LEAST 50 PER CENT GREATER THAN THE WALL THICKNESS REQUIRED BY TABLE 2. THE MAXIMUM ALLOWABLE WORKING PRESSURES FOR COPPER TUBES USED IN WATER-TUBE BOILERS SHALL BE FOR THE VARIOUS DIAMETERS AND GAGES MEASURED BY BIRMINGHAM WIRE GAGE AS GIVEN IN TABLE 2 1/2, BUT NOT TO BE USED FOR PRESSURES TO EXCEED 250 LB. COPPER TUBES SHALL NOT BE USED WITH SUPERHEATED STEAM.

TABLE 2 1/2 MAXIMUM ALLOWABLE WORKING PRESSURES FOR COPPER TUBES FOR WATER-TUBE BOILERS

For Different Diameters and Gages of Tubes

Outside diam. of tube, in. D	Gage—B. W. G. 12	11	10	9	8	7	6	5	4
2	170	231	250	250	250	250	250	250	250
3 1/4	...	...	101	142	215	250	250	250	250
4	...	...	...	...	128	173	242	250	250
5	...	...	...	...	...	...	143	184	218

$P = \left(\frac{t - 0.039}{D}\right)12000 - 250$

Where P = Maximum allowable working pressure, lb. per sq. in.
t = Thickness of tube wall, in.
D = Outside diameter of tube, in.

PAR. 185 ADD THE FOLLOWING TO THE REVISED FORM PRINTED IN THE DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

WHERE PLATES ARE PLANED OR MILLED DOWN IT SHALL BE FOR THE ENTIRE CIRCUMFERENCE OF THE JOINT, AND THE FILLET AT THE EDGE OF THE PLANING SHALL BE NOT LESS THAN 1 IN. RADIUS.

PAR. 194 REVISE PARAGRAPH AS PRINTED IN AUGUST 1922 ISSUE OF MECHANICAL ENGINEERING TO READ AS FOLLOWS:

194 The longitudinal joint of a dome 24 in. or over in inside diameter shall be of butt and double-strap construction or made without a seam of one piece of steel pressed into shape, and its flange shall be double-riveted to the boiler shell. IN THE CASE OF

NOTE:—Matter in caps.—added matter; Matter in smaller type in brackets—to be deleted.

A DOME LESS THAN 24 IN. IN DIAMETER, FOR WHICH THE PRODUCT OF THE INSIDE DIAMETER AND THE MAXIMUM ALLOWABLE WORKING PRESSURE DOES NOT EXCEED 4000 IN. LB., ITS FLANGE MAY BE SINGLE RIVETED TO THE BOILER SHELL AND THE LONGITUDINAL JOINT MAY BE OF THE LAP TYPE PROVIDED IT IS COMPUTED WITH A FACTOR OF SAFETY NOT LESS THAN 8.

WHEN A DOME IS LOCATED ON THE BARREL OF A LOCOMOTIVE-TYPE BOILER OR ON THE SHELL OF A HORIZONTAL-RETURN-TUBULAR BOILER, THE DIAMETER OF THE DOME SHALL NOT EXCEED SIX-TENTHS THE DIAMETER OF THE SHELL OR BARREL OF THE BOILER.

ALL DOMES SHALL BE SO ATTACHED THAT ANY WATER WHICH ENTERS THE DOME ALONG WITH THE STEAM CAN DRAIN BACK INTO THE BOILER.

FLANGES OF DOMES SHALL BE FORMED WITH A CORNER RADIUS, MEASURED ON THE INSIDE, OF AT LEAST TWICE THE THICKNESS OF THE PLATE FOR PLATES 1 IN. THICK OR LESS, AND AT LEAST THREE TIMES THE THICKNESS OF THE PLATE FOR PLATES OVER 1 IN. IN THICKNESS.

WHEN BOILER SHELLS ARE CUT TO APPLY STEAM DOMES, THE NET AREA OF METAL, AFTER RIVET HOLES ARE DEDUCTED, IN FLANGE AND LINER, IF USED, MUST NOT BE LESS THAN THE AREA REQUIRED BY THESE RULES FOR A LENGTH OF BOILER SHELL EQUAL TO THE LENGTH REMOVED. THE HEIGHT OF VERTICAL FLANGE EQUAL TO THREE TIMES THE THICKNESS OF THE FLANGE SHALL BE INCLUDED IN THE AREA OF THE FLANGE (SEE PARS. 187 AND 188).

PAR. 195 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Insert the word "unstayed" before the word "dished," the first word of the fourth section of this paragraph.

PAR. 212*a* NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Omit the words "except handholes" in the 9th and 16th lines.

PAR. 216 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Change the figure "1.25" at the beginning of line 11, to "1.5."

PAR. 218 CHANGE THE FIRST SENTENCE OF REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING AS FOLLOWS:

218 When STAYS ARE REQUIRED the portion of the heads below the tubes in a horizontal-return-tubular boiler SHALL BE SUPPORTED AT THE FRONT HEAD BY THROUGH STAYS WITH NUTS INSIDE AND OUTSIDE AND AT THE REAR HEAD BY ATTACHMENTS WHICH DISTRIBUTE THE STRESS.

PAR. 230 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Insert the word "determined" after the word "length" in the 9th line.

PAR. 231 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Insert the words "or rivets" after the word "staybolts" in the 16th line.

PAR. 239 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Add the words "or flame" to section a-4.

PAR. 247 REPLACE THE REVISED FORM PRINTED IN JULY 1922 ISSUE OF MECHANICAL ENGINEERING WITH THE FOLLOWING:

247 Where NO RULES ARE GIVEN and it is impossible to calculate with a reasonable degree of accuracy the strength of a boiler structure or any part thereof, a full-sized sample shall be built by the manufacturer and tested IN A MANNER TO BE PRESCRIBED BY THE BOILER CODE COMMITTEE AND in the presence of or more representatives appointed to witness such test.

PAR. 253 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Revise fifth line of the first section to read: DIAMETER FOR [thicker] MATERIAL [not] MORE THAN $^5/_{16}$ [$^5/_8$] IN. THICK.

Insert the word "such" before the first word "holes" in the second section.

PAR. 260 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Omit the words "when used" in the first line.

In Fig. $20^1/_2$, turn this figure upside down and cross-hatch and revise so that the cross-section of flanged manhole frame will have a curved line representing the lower edge of plate around hole.

PAR. 261 REPLACE THE REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING WITH THE FOLLOWING:

261 The strength of the rivets in shear on each side of a [manhole] frame or ring reinforcing MANHOLES OR OTHER OPENINGS SUCH AS THOSE CUT FOR STEEL NOZZLES AND BOILER FLANGES OVER 3 IN. PIPE SIZE shall be at least equal to the tensile strength of the maximum amount of the shell plate removed by the opening and rivet holes for the reinforcement on any line parallel to the longitudinal axis of the shell, through the manhole, or other opening.

PAR. 268 REVISE FIRST SECTION OF FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING TO READ AS FOLLOWS:

268 *Threaded Openings.* ALL PIPE THREADS SHALL CONFORM TO THE AMERICAN PIPE STANDARD AND ALL connections 1 in. PIPE SIZE or over shall have not less than the number of threads given in Table 8. FOR SMALLER PIPE CONNECTIONS THERE SHALL BE AT LEAST FOUR THREADS IN THE OPENING.

PAR. 269 REVISE FIRST SENTENCE OF FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING TO READ AS FOLLOWS:

269 *Safety Valve Requirements.* Each boiler HAVING MORE THAN 500 SQ. FT. OF WATER HEATING SURFACE OR IN WHICH THE GENERATING CAPACITY EXCEEDS 2000 LB. PER HOUR, shall have two or more safety valves.

PAR. 273 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Replace the word "body" in the 4th and 5th lines with the word "casing."

PAR. 274 REVISE FIRST SECTION OF FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING AS FOLLOWS:

274 The MINIMUM AGGREGATE [total] relieving capacity of ALL OF the safety valve or valves required in a boiler shall be not less than that determined on the basis of 6 lb. of steam per hour per sq. ft. of boiler heating surface for water-tube boilers. For all other types of power boilers, the total relieving capacity shall be not less than that determined on the basis of 5 lb. of steam per hour per sq. ft. of boiler heating surface, for boilers with maximum allowable working pressure above 100 lb. and on the basis of 3 lb. of steam per hour per sq. ft. of boiler heating surface for boilers with maximum allowable working pressures at or below 100 lb. per sq. in. IN MANY CASES A GREATER RELIEVING CAPACITY OF SAFETY VALVES WILL HAVE TO BE PROVIDED THAN THE MINIMUM SPECIFIED BY THIS RULE AND IN EVERY CASE THE REQUIREMENT OF PAR. 270 SHALL HOLD.

REVISE ADDED MATTER AT END OF SECOND SECTION AS FOLLOWS:

WHERE THE OPERATING CONDITIONS ARE CHANGED, THE SAFETY VALVE CAPACITY SHALL BE INCREASED, IF NECESSARY, TO MEET THE NEW CONDITIONS AND BE IN ACCORDANCE WITH PAR. 270.

PAR. 278 REVISE ADDED MATTER OF FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING AS FOLLOWS:

EACH VALVE SHALL HAVE AN OPEN DRAIN THROUGH THE CASING BELOW THE LEVEL OF THE VALVE SEAT. FOR VALVES EXCEEDING 2 IN. PIPE SIZE, THE HOLES SHALL BE TAPPED FOR DRIP PIPE. IN THE CASE OF FIRE-TUBE BOILERS, THE BOILER OPENINGS FOR SAFETY VALVES SHALL BE NOT LESS THAN THOSE CORRESPONDING TO AN EVAPORATION OF 5 LB. OF STEAM PER HOUR PER SQ. FT. OF HEATING SURFACE AND TO SAFETY VALVES HAVING THE INTERMEDIATE LIFTS AND CORRESPONDING RELIEVING CAPACITIES GIVEN IN TABLE 15.

PAR. 280 REVISE LAST LINE OF FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING AS FOLLOWS:

DIAMETERS of [all of] the safety VALVE CONNECTIONS AND SHALL ALSO MEET THE REQUIREMENTS OF PAR. 278.

PAR. 281 REVISE SECOND SENTENCE OF FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING AS FOLLOWS:

To close after blowing down not more than 4 PER CENT OF THE SET PRESSURE, EXCEPT FOR WORKING PRESSURES OF 50 LB. PER SQ. IN. OR LESS, IN WHICH CASE THE BLOW DOWN SHALL NOT EXCEED 2 LB. PER SQ. IN.

PAR. 287 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Replace the word "body" in the first line, with the word "casing."

PAR. 289 REVISE SECOND LINE ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING AS FOLLOWS:

heated steam, shall have a CASING OF STEEL, STEEL ALLOY, OR EQUIVALENT HEAT-RESISTING MATERIAL, INCLUDING ALL PARTS WHICH

INSERT THE WORDS "(see Par. 12)" after the word "OUTLET" at the end of the third line, ending the sentence. Begin new sentence with the words THE VALVE SHALL HAVE A FLANGED, etc.

PAR. 321 ADD THE FOLLOWING SENTENCE TO FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

FOR STEAM PRESSURES OF 200 LB. AND OVER, THE WATER CONNECTIONS SHALL BE OF STEEL PIPE OR TUBING, WROUGHT-IRON PIPE OR THE EQUIVALENT.

PAR. 324 NOTE THE FOLLOWING ON REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING:

Insert the word "or" before the word "over" in the second line.

PAR. 332 REPLACE THE REVISED FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING WITH THE FOLLOWING:

332 Each boiler shall conform in every detail to these Rules, and shall be distinctly stamped with the symbol as shown in Fig. 23, denoting that the boiler was constructed in accordance therewith.

After obtaining the stamp to be used when boilers are to be constructed to conform with the A.S.M.E. Boiler Code, a state inspector, municipal inspector, or an inspector employed regularly by an insurance company which is authorized to do a boiler insurance business in the state in which the boiler is built and in the state in which it is to be used, if known, is to be notified that an inspection is to be made and he shall inspect such boilers during construction and after completion. At least two inspections shall be made, one before reaming rivet holes and one at the hydrostatic test. In stamping the boiler after completion, if built in compliance with the code, the builder shall stamp the boiler in the presence of the inspector, after the hydrostatic test, with the A.S.M.E. Code stamp, the builder's name and the serial number of the manufacturer. A data sheet shall be filled out and signed by the manufacturer and the inspector. This data sheet together with the stamp on the boiler shall denote that it was constructed in accordance with the A.S.M.E. Boiler Code.

IN CASES WHERE BOILERS CANNOT BE COMPLETED AND HYDROSTATICALLY TESTED BEFORE SHIPMENT, THE PROPER STAMPINGS SHALL BE APPLIED AT THE SHOP AND TWO DATA SHEETS SIGNED AS HEREIN PROVIDED BY THE SAME OR DIFFERENT INSPECTORS COVERING THE PORTIONS OF THE INSPECTIONS MADE AT THE SHOP AND IN THE FIELD, THE DATA SHEETS EACH TO BE SEPARATELY SENT TO THE PROPER DESTINATION.

Each boiler shall be stamped adjacent to the symbol as shown in Fig. 24, with the following items with intervals of about one-half inch between the lines:

1 REGISTERED NUMBER
2 A.S.M.E. [Manufacturer's] serial number WHICH MAY BE THE MANUFACTURER'S SERIAL NUMBER.
[2 State in which boiler is to be used.]
[3 Manufacturer's State standard number.]
3 Name of manufacturer.
4 Maximum working pressure when built.
5 WATER HEATING SURFACE IN SQ. FT.
[5 State's number.]
6 Year put in service.

Items 1, 2, 3, 4, and 5 to be stamped at the shop where built.

Item 6 is to be stamped by the proper authority at point of installation.

PORTABLE BOILERS OF 100 HP. OR LESS SHALL HAVE THE STAMPINGS AS HEREIN PROVIDED APPLIED ON A NON-FERROUS PLATE 3 IN. X 4 IN. SIZE WHICH SHALL BE, AS NEARLY AS PRACTICABLE, IRREMOVABLY FASTENED TO THE BOILER NEAR THE WATER-COLUMN CONNECTIONS. ALL OTHER BOILERS MAY HAVE THE STAMPINGS SO APPLIED OR THEY MAY BE APPLIED DIRECTLY TO THE BOILER STRUCTURE.

MANUFACTURERS, BEFORE USING THE A.S.M.E. SYMBOL, SHALL BE GIVEN A REGISTRATION NUMBER BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. THIS NUMBER SHALL BE STAMPED DIRECTLY ON THE BOILER OR PLACED ON A NON-FERROUS PLATE. THE REGISTRATION NUMBER SHALL BE PLACED NOT MORE THAN ONE-HALF INCH ABOVE THE SYMBOL AND CENTERING ON IT. INSPECTION JURISDICTIONS SHALL BE FURNISHED BY THE A.S.M.E. A LIST OF REGISTERED MANUFACTURERS, WITH THE CORRESPONDING REGISTRATION NUMBERS ISSUED.

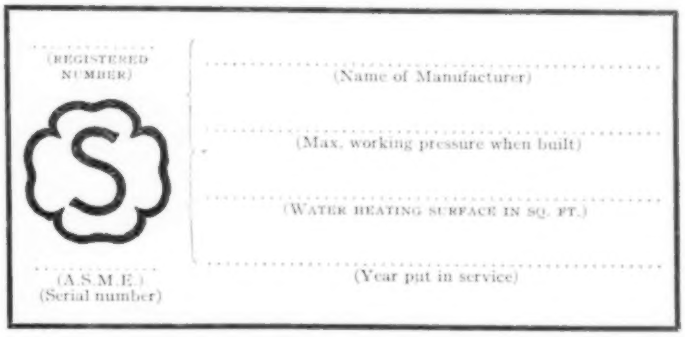


FIG. 24 FORM OF STAMPING

PAR. 429 OMIT REVISION IN FORM PRINTED IN DECEMBER 1922 ISSUE OF MECHANICAL ENGINEERING.

New Revisions of A.S.M.E. Boiler Code, 1923

PAR. 22 REVISED:

22 *Tubes for Fire-Tube Boilers.* The minimum thickness of tubes used in fire-tube boilers measured by Birmingham wire gage, for maximum allowable working pressures not exceeding 175 lb. per sq. in., shall be as follows:

	Gage—B. W. G.	
	Steel or Wrought Iron	COPPER
Diameters 1 in. or over, but less than $2^1/_2$ in.	13	10
Diameter $2^1/_2$ in. or over, but less than $3^1/_4$ in.	12	8
Diameter $3^1/_4$ in. or over, but less than 4 in.	11	7
Diameter 4 in. or over, but less than 5 in.	10	6
Diameter 5 in.	9	5

For each increase of one gage in thickness above that shown in the table, the maximum allowable working pressure will be increased by 200 lb. divided by the diameter of the tube in inches.

COPPER TUBES SHALL NOT BE USED FOR PRESSURES IN EXCESS OF 250 LB. PER SQ. IN.

COPPER-TUBE SPECIFICATIONS: SPECIFICATIONS FOR SEAMLESS COPPER BOILER TUBES WILL BE INCLUDED FOLLOWING THOSE OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION B 13–18.

PAR. 179 REVISED:

179 *Maximum Allowable Working Pressure.* The maximum allowable working pressure is that (at which a boiler may be operated as) determined by employing the factors of safety, stresses, and dimensions designated in these Rules.

(Remainder of paragraph unchanged)

PAR. 333*c* REVISED:

c On traction, portable or stationary boilers of the locomotive type or Star water-tube boilers—on the furnace end, above the handhole. OR ON TRACTION BOILERS OF THE LOCOMOTIVE TYPE—ON THE LEFT WRAPPER SHEET FORWARD OF THE DRIVING WHEEL.

(*Continued on page 273*)

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities Papers and Proceedings of

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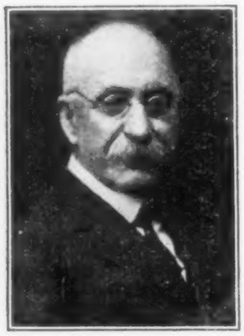
Plain Speaking



DR. IRA N. HOLLIS

"BROTHERHOOD of all engineers and their united action in any service that will be for the good of our country." This motto, worthy and characteristic of Dr. Ira N. Hollis, the 1917 President of The American Society of Mechanical Engineers, appeared in the volume of Transactions for that year, during which he took his place as first chairman of the Engineering Council, an important first step toward the national coöperation of engineering societies. Dr. Hollis' life has been given to the ideal of service: he realizes the need for straightforward engineering thought in our modern civilization, and he has confidence in the ability of the engineering profession to perceive this need and adapt itself for it. However, he believes that facts must be faced, and in a recent communication he directs our attention to the purposes of The Federated American Engineering Societies. We are privileged to print below a portion of this letter. Read it carefully.

"The plain truth is that the engineers have not yet come to an understanding of themselves. In their own minds they have not yet passed out of that classification that the public has given them, namely, of being technical men called in to advise only on technical questions. A few of them have passed out of that category in the esteem of the public, but the rank and file are still laboring under the disadvantage of being simply hired men. One may well wonder how long this is to continue. We know perfectly well that we all ought to be good citizens and that we ought to have our share in the government of nation and city. That does not at all mean neglecting our professions, but it does mean giving some time and some money toward wielding our own proper influence.

"The Federated American Engineering Societies was organized as a kind of senate wherein would appear public questions. It sprang out of the old Engineering Council where the four oldest national societies got together on certain things that are common to all in order that they might not have to present a too great variety of opinions or decisions. The old Council was really the outcome of an effort to work together. This new society, which takes in local and national societies alike, holds out a prospect of bringing the engineer before the public as nothing else ever has. If it did absolutely nothing else than to teach the American people the value of the engineering profession and capacity of its members to assist in the decision of great national problems, it would justify itself. The main difficulty is found in its support. Few engineers are opposed to it in principle, but there are altogether too many who do not care to put in even the small amount of dues now required. Some societies find the total amount paid over by them a burden and fear to lose their own members by diverting any part of their income to this activity. It is unenlightened in men to demand immediate personal benefit from the payment of a small amount, $1.00 or $1.50 per year, for the support of this public institution. They might as well ask for some immediate personal benefit springing out of their taxes paid over to the state and Government. We all get a benefit from the taxes, but not all alike. The man who pays a thousand dollars gets only just what the man who pays ten dollars gets. We pay willingly because our taxes mean the maintenance of society. The case of the F.A.E.S. payment is the same as any public assessment. The engineer has been the hired man of the ages. He is found from time to time in the historic period. Upon him the Egyptians depended for the life of the population in the Nile Valley. The same is true in the valley of the Euphrates and the Tigris. Wherever he has been absent, the nation has gone steadily down hill, and yet he has, up to within a comparatively recent period, had little consideration from the general public simply because he has had a tendency to treat himself as purely a technical man. There is only one way to recover from that in our modern days, and that is by organization. We know it. There must be a thousand other associations or organizations in the United States for different purposes, and we cannot afford to become ineffective in that direction. The Federated Societies, or something like them, offer us the only means of making the profession well known in its larger possibilities."

Safe Engineering Versus Safety Engineering

Champlain Studios, N. Y.
WM. J. VENNING

EVER SINCE the earliest ages man has been forced to provide means for self-protection. Primitive man fashioned cudgels from stone which he used as weapons of defense as well as offense, and for aid in self-preservation lived in groups with his fellow-beings. As these groups grew in size and men were compelled to depend upon one another for protection of the group as a whole, it became necessary to make laws defining the conduct of the individual. Some of these laws the present-day safety engineer would call strictly "safety-first" laws. For instance, in the Bible, the eighth verse of the 22nd chapter of Deuteronomy specifically interprets a building law; then again King Hamurabi, whose death occurred in Babylon in the year of 2185 B.C., promulgated a building law as follows: "If a builder build a house for a man and do not make its construction firm, that builder shall be put to death." And so on down through the ages to our present safety codes.

The need for a safety code is usually determined by a study of accident causes, and since industrial development has been so rapid, what was considered a good code yesterday is obsolete today. Then again, the lack of uniformity and agreement between the various states has always been an insurmountable obstacle.

"Safe Engineering" takes into consideration not only safety factors in design and construction but should also provide for safe operation or use. "Safety Engineering" is applied where design and construction have failed to provide for safety operation or use. That it would be much cheaper to always incorporate safe engineering in construction and design cannot be gainsaid. Fifty machines of the same design, made by the same builder, sold to fifty different users, if not properly guarded during construction, will eventually be guarded fifty different ways by the purchasers with varying degrees of efficiency and cost strictly in accordance with the purchaser's state of mind. Where it has been the writer's task to initiate the uninformed industrial plant owner into the realm

of safety by offering certain recommendations for safeguards as a result of plant inspection, one of the first questions asked is, "Why doesn't the manufacturer properly guard the machine when he constructs it?"

There are three important points to be considered in the safeguarding of machinery:

1 Drive belts and pulleys or drive device
2 Moving and reciprocating parts
3 Point of operation.

The drive belts or device of any machine are easily guarded, the most efficient construction being of expanded metal, perforated metal, or woven wire, mounted on an angle-iron frame of the box type and securely fastened in place. Guards of this type will stand hard usage and repay for their cost in reduced insurance rates.

To guard the moving and reciprocating parts invariably requires a little more thought. Solid guards made of sheet iron are usually employed, although for quantity production on similar-type machines a cast guard made from a pattern is the least expensive. The installation of these guards must permit of easy machine adjustment and in no way interfere with efficient operation.

The point of operation of a machine is that part where the stock is fed in. Automatic feeds with the moving parts guarded and which remove the hazards inherent in hand feeding are most desirable. Where hand feeding is necessary, the guard should fully protect the operator and expedite the work. Although the machine designer and builder can easily guard the drive device and moving parts of any machine during initial construction and can do it much cheaper than the purchaser because the work is done on a production basis, he cannot always guard the point of operation because of the many and various kinds of work performed on the same machine; for instance, a gate guard that would be practical on a punch press for second-operation hand-fed work, would not do for blanking work from strip stock. Particularly in small shops with limited mechanical equipment does the safety engineer meet with this difficult problem. However, closer coöperation between the machine builder and the professional safety engineer should gradually solve this problem, and teaching safety engineering in the technical schools and engineering colleges should eventually make "Safe Engineering" the principal factor in design and construction.

In every instance sound engineering practice should be followed to obtain the best results.

An efficiently guarded machine will give the operator a greater sense of security, thereby removing the necessity of continual caution.

Guarding prevents property damage often caused by careless handling of stock around and near machinery in motion.

Well-guarded machines permit the piling of stock close up to the machine and conserve floor space.

A machine properly guarded makes it possible to use unskilled labor on many operations that would otherwise call for skilled labor, because part of the skill or training consists in being able to localize and avoid the hazard.

Efficient machine guarding reduces labor turnover by keeping the worker on the job uninjured, thereby increasing production through increased shop efficiency.

William J. Venning.[1]

Asiatic Markets for Industrial Machinery

For years there has been a not unfounded complaint to the effect that the American exporter does not receive the same amount of support from his government as do his competitors from theirs. In particular, attention was called to the excellent information service maintained for the export trade before the war by the British and German governments.

That this situation is rapidly changing for the better is well indicated, for example, by a recent publication of the Bureau of Foreign and Domestic Commerce of the Department of Commerce, entitled Asiatic Markets for Industrial Machinery, by Walter H. Rastall, Mem. A.S.M.E.

This book is a veritable repository of information not only on the actual export statistics but on conditions surrounding the business of exporters of industrial machinery to Asiatic markets, their purchasing power and requirements. The investigation deals primarily with machinery used in and about factories and power plants, but also includes data on construction machinery, mining machinery and certain other types—not, however, such special machinery as cash registers, automobiles and sewing machines. In addition to the more obvious features of such an investigation, an effort has been made to examine into the advertising, financing, and selling methods employed by foreign competitors as well as by successful American exporters, and certain sections of the report have been prepared with the idea of furnishing detailed information needed by men who are starting in export work as well as those with more ample experience.

In the letter of submittal of the Director of the Bureau to Secretary of Commerce Hoover, it is pointed out that a notable movement toward industrialization is in progress in the more important Asiatic countries, and Mr. Rastall is decidedly optimistic with regard to the prospects for development. Considering that the total population of the countries covered by this report is probably well in excess of half a billion souls and that they have large natural resources still awaiting development, the subject of these markets for American machinery is one well worthy of serious attention.

[1] President, American Society of Safety Engineers.

Montreal Spring Meeting

The coming Montreal Spring Meeting of The American Society of Mechanical Engineers is the first to be held outside of the borders of the country since 1894, when one was also held in Montreal. It is hoped that the coming convention, **May 28–31,** will afford an excellent opportunity for the development of the already amicable relations between the engineers of Canada and the United States. The members of the Engineering Institute of Canada have been invited to attend, and the Montreal Branch of the Canadian Institute is coöperating wholeheartedly in the plans for the meeting.

The technical program will of course stress Hydroelectric Development, and two sessions will be devoted to that subject. The other topics to be considered are Management Engineering, with special application to the Paper and Pulp Industry, Port Development, in which the harbor problems of Montreal will be treated; Textiles; Railroads; and Fuels. The Professional Divisions on Power, Management, Materials Handling, Textiles, Railroads, and Fuels are coöperating in these sessions and the Divisions on Aeronautics, Forest Products, and Machine Shop Practice are presenting papers for the miscellaneous sessions.

The meeting will be notable for its entertainment features which will comprise a smoking concert at which members of the A.S.M.E. will be the guests of the Montreal branch of the Engineering Institute of Canada, a dinner dance, and excursions to points of scenic and engineering interest in and around Montreal. The number of points to be visited make Montreal an exceptionally attractive convention city. Following the meeting a trip to Grandmere and Quebec is being planned.

Detailed information will be found in the current issues of *A.S.M.E. News* and the tentative program for the meeting will appear in the April 7 issue.

Bureau of Standards Report on Welded Vessels to Come Later

The complete report on the Bureau of Standards investigations on welded pressure vessels which was promised in the March issue of Mechanical Engineering, page 210, has not as yet been received by the A.S.M.E. Boiler Code Committee and its presentation will necessarily be deferred to a later issue. It is hoped that it will be possible to present a comprehensive report on this most interesting and valuable series of tests.

Erratum

In extracts from the U.E.S. treasurer's report for 1922, printed in the March issue of Mechanical Engineering, page 205, the cash on hand as of December 31, 1922 was inadvertently stated as amounting to $138,294.27. The figure should have been $32,754.89.

Dr. M. A. Hunter Presents Facts About High-Temperature Alloys at Newark Meeting

The requirements of electric heaters, valves for internal-combustion engines, containers for high-temperature furnace work and parts of steam boilers for use with high superheats have made the subject of high-temperature alloys of great interest to mechanical engineers. The Metropolitan Section of the A.S.M.E. held a meeting on this subject in Newark on February 13 which was addressed by Dr. M. A. Hunter, professor of electrochemistry at Rensselaer Polytechnic Institute. Dr. Hunter presented many facts which are of importance to those engaged in work of the nature mentioned.

The line between high- and low-temperature alloys may be roughly drawn at around 1800 deg. fahr., although not all metals or alloys melting above this temperature can be used, either for technical or commercial reasons. Thus the metals of the platinum groups are too rare and costly, at least for large-scale applications, although used extensively for high-temperature chemical work and pyrometry. Metals like tungsten, tantalum, and molybdenum are chemically unstable when exposed to air, but can be used as in gas-filled electric lamps at extremely high temperature under proper conditions.

The most common metals available for high-temperature commercial work in air or active gases are iron, melting at 1520 deg. cent.; chromium, at 1550 deg.; cobalt, at 1590 deg.; nickel, at 1450 deg.; silicon, at 1420 deg.; manganese, at 1225 deg.; and copper, at 1084 deg. These are the chief constituents of high-temperature alloys, which, however, may contain other metals, such as tungsten, in small quantities.

In general, high-temperature alloys consist of two or three components united in what metallurgists call a "solid solution."

High-temperature alloys may be subjected to the action of high temperature alone or in combination with chemical actions—of which the most important are those of a corrosive nature—and mechanical stresses, and alloys good for one purpose may not be suitable for another. For temperatures below 700 deg. cent., particularly where chemical attack is to be feared (except in the case of nitric acid), copper-nickel mixtures form an important class. Of this the best known is monel metal, an alloy of about 30 per cent copper and 70 per cent nickel. Its melting point is said to be 2480 deg. fahr. Monel metal may be used in the form of castings, rolled sheets, and drawn wire. Its strength approaches that of mild steel in corresponding products. At temperatures above 700 deg. cent. it is apt to oxidize. Cast monel metal usually has a high silicon and low manganese content, as this helps the fluidity of the metal; rolled or drawn monel is low in silicon and high in manganese.

An alloy consisting of 40 per cent nickel and 60 per cent copper, known as "Advance wire," is of considerable value in temperature-measurement work as it has a zero temperature coefficient over a range of 300 deg.

Some of the valuable high-temperature alloys are characterized by the presence of chromium in considerable degree. The earliest of those developed commercially were the nickel-chrome alloys which have found extensive application as electrical resistance wire and heating elements for electric stoves and the like. The most used combination is 80 per cent nickel and 20 per cent chromium. This alloy is noteworthy chiefly for its ability to withstand oxidation at high temperatures. Alloys with practically any proportion of chromium can be formed by casting, but where the metal has to be worked the percentage of chromium should not exceed 20 per cent. The matter of carbon content has to be watched in this alloy with extreme care. The nickel-chromium wire patents are understood to have expired in February of the current year.

Alloys of nickel, iron, and chromium are used for applications where temperatures not exceeding 900 deg. cent. prevail. The alloy nichrome (containing 26 per cent iron, 12 per cent chromium, the remainder nickel and impurities) is used as rod, strip, or wire in electrical resistors. As a cast material it finds increasing applications in carbonizing boxes, retorts, lead and cyanide pots, etc. These alloys do not carburize at high temperatures, crack, warp, or scale, and except under high sulphur conditions in the fuel used, give very satisfactory service. One of their remarkable properties is the ability to retain a considerable proportion of their original strength at 1000 deg. cent., which is very different from the behavior of steel and similar alloys.

Nichrome exhibits great tensile strength at high temperatures. At 1000 deg. cent., material which at room temperature shows 67,000 lb. strength has 12,000 lb. strength, an elongation of 55 per cent, and a reduction in area of 15 per cent. This same metal at 850 deg. cent. has an ultimate strength of 20,000 lb., and at 600 deg. cent. 44,000 lb. It has been successfully used for internal-combustion-engine valves.

Other high-temperature alloys deserving consideration for commercial work are those of the chrome-iron group. These contain chromium in varying percentages from about 20 up to 35 or even 40 and are remarkable for their high resistance both to temperature and to chemical stresses. The physical properties of the chrome-iron alloys (which, of course, differ basically from cast nichrome in that they contain no nickel) are essentially determined by their carbon content, alloys with carbon content below 0.30 per cent machining practically as freely as cold-rolled steel; with carbon content between 0.30 to 1.5 the machining becomes increasingly difficult, and finally, when the 1.5 per cent carbon limit has been reached a material is obtained which casts very well but cannot be machined by ordinary means.

In the discussion which followed, the question was raised as to whether these high-temperature alloys would withstand the combination of temperatures up to 1200 deg. fahr. with pressures such as are encountered, for example, in oil-cracking stills (Burton and Rittman processes). In this connection, it might be mentioned that in France an alloy has been developed of practically the same composition as cast nichrome but with the addition of about 2 per cent tungsten, and that tubes cast of this material have been used in the Claude ammonia process where both high temperatures and high pressures are used simultaneously.

David A. Decrow Dies

David A. Decrow, manager of the water-works department of the Worthington Pump & Machinery Corporation, New York City, died on February 15, 1923. Mr. Decrow was born in Bangor, Me., in 1858, where he received his early education. He was graduated from the University of Maine, College of Mechanic Arts, with the class of 1879.

In the early eighties he became associated with the Holly Manufacturing Co., Lockport, N. Y. His promotions with this company were rapid; in 1893 he was made designing engineer, in 1900 chief engineer, and in 1903 secretary of the firm.

Some years ago the Holly Manufacturing Co. was combined with the Snow Steam Works at Buffalo, N. Y., as part of the International Steam Pump Co., and at that time Mr. Decrow went to Buffalo to take charge of the pumping machinery manufactured by both companies. In April, 1916, the International Steam Pump Co. was succeeded by the Worthington Pump & Machinery Corporation. Soon after this Mr. Decrow was called to the New York office to become manager of the water-works department, the position he held until his death.

Mr. Decrow was active as a member and recently as chairman of the A.S.M.E. Committee on the Test Code for Reciprocating Displacement Pumps. He was a member of a number of clubs and organizations in Buffalo.

Engineering Division, National Research Council, Elects Officers

Dr. F. B. Jewett, vice-president of the Western Electric Company, New York N. Y., has been elected chairman of the Engineering Division of the National Research Council, to succeed Alfred D. Flinn, who resigned that office last year to become director of Engineering Foundation.

E. B. Craft, chief engineer of the Western Electric Company, was elected vice-chairman of the Division, and M. Holland, formerly of the U. S. Air Service, McCook Field, Dayton, Ohio, appointed director. G. H. Clevenger was reëlected a vice-chairman and W. Spraragen secretary.

Philadelphia Holds Successful Machine Meeting

Technical, Historical, and Economic Phases of Machine Tools and Machine-Shop Practice Discussed Before Large Gatherings. Dean Dexter S. Kimball, President John Lyle Harrington of A.S.M.E., and E. F. DuBrul among the Speakers

TECHNICAL, historical, and economic phases of the machine-tool industry and machine-shop practice were presented and discussed at the Conference on Machine Tools which was held in Philadelphia on February 27 under the auspices of the Engineers' Club of Philadelphia and the Philadelphia Sections of The American Society of Mechanical Engineers and the American Institute of Electrical Engineers, and with the coöperation of the Machine Shop Practice Division of The American Society of Mechanical Engineers.

Over three hundred attended the sessions and dinner, and the good papers, interesting discussion, and increased fellowship among the engineers in machine-shop practice which resulted from the meeting form an achievement of which the Philadelphia engineers may well be proud.

Two sessions were held, the afternoon session being given over to the presentation and discussion of technical papers, the first of which, on the Effect of Variations in Design of Milling Cutters on Power Requirements and Capacity, by Prof. James A. Hall, of Brown University, and Benjamin P. Graves, of the Brown & Sharpe Mfg. Co., appeared in the March issue of MECHANICAL ENGINEERING. The discussion is given later in this account of the meeting. The second paper, on the Design and Construction of Large Machine Tools, by George H. Benzon, Jr., of William Sellers & Co., appears as the leading article of this issue. Charles Penrose, chairman of the Philadelphia Section of the A.S.M.E., was introduced as presiding officer by Dr. Robert H. Fernald, president of the Engineers' Club of Philadelphia. Mr. Penrose called on Frank O. Hoagland, of Worcester, Mass., chairman of the Machine Shop Practice Division of the A.S.M.E., who told of the plans of his Division in meetings, research, and standardization.

Dinner was served at the Engineers' Club and was followed by a brief talk by President John Lyle Harrington of The American Society of Mechanical Engineers, who outlined the steps in the development of modern equipment for industry and transportation. Dean Dexter S. Kimball of Cornell University, past-president of the A.S.M.E., followed with a lantern-slide talk on the History of the Machine Tool and Its Effect on Present-Day Civilization. He elaborated somewhat on the paper which appeared in the March issue of MECHANICAL ENGINEERING and his picturesque portrayal of the romance of mechanical development held the interest of those who filled the seats of the auditorium and a large number who stood throughout his talk. He was followed by Ernest F. DuBrul, general manager of the National Machine Tool Builders' Association, who spoke on the Economic Features of the Machine-Tool Industry. Mr. DuBrul's paper appeared as the leading article in the March issue of MECHANICAL ENGINEERING. Ross B. Mateer, secretary of the Philadelphia Section of the A.I.E.E., presided at this evening session.

Discussion at Afternoon Session

Following the presentation of the paper on Effect of Variations in Design of Milling Cutters on Power Requirements and Capacity by Professor Hall and Mr. Graves, written discussion by Fred A. Parsons[1] was presented. Mr. Parsons took exception to the use of the terms "feed per tooth" or "maximum chip thickness" as bases for the comparison of cutting tools. He favored "average thickness of chip" as a satisfactory basis as it eliminated differences in depth of cut, diameter of cutter, and type of tool. He agreed with the conclusions of the authors in regard to the superiority of coarse-tooth cutters for general purposes.

W. A. Knight[2] stated as his opinion that spiral face milling for large flat-surface work should be obsolete as the cutting edge of a milling cutter must ride the surface every time a cut is taken and the cut is comparatively short.

[1] Kearney & Trecker Corp., Milwaukee, Wis. Mem. A.S.M.E.

[2] Acting-Director Industrial Arts Dept., Ohio State University. Mem. A.S.M.E.

R. Poliakoff[1] submitted a written discussion descriptive of a milling-machine dynamometer. He also pointed out that the end thrust found when using spiral cutters could be overcome by making the cutters with one-half having a right-hand spiral and the other half a left-hand spiral. He agreed with the authors that a 10-in. rake angle is desirable.

Carl G. Barth[2] agreed with the findings of the authors in the advantages of milling cutters with few teeth. He expressed the desire for further experimentation to discover the value of the exponent n in the formula P = constant $\times\, wf^n$ for the pressure on the lathe slicing or cutting-off tool taking a cut w with a feed f. Mr. Barth gave his derivation of an expression for horsepower consumption per cubic inch of metal removed with milling cutters as follows:

$$\mathrm{HP}_c = \text{constant}\ (N/F)^{1-n}\ (D/H)^{1/4}$$

in which N = number of teeth in cutter, F = feed per revolution, H = depth of cut and D = diameter of cutter, and which thus shows that HP_c increases with the number of teeth and decreases with feed per revolution, both raised to the power $(1-n)$; and increases with diameter of cutter and decreases with depth of cut, both raised to the power $^1/_4$; and that this is because the increased diameter reduces while increased depth of cut increases the chip thickness.

John Airey[3] presented a lengthy oral discussion in which he decried the fact that so little effort had been put into research work in machine-shop practice. He agreed with the authors on the subject of rake and lack of consistency of material. He disagreed in the manner of measuring the power consumption and in ascribing some parts of the power loss to the cutter. He gave some of the information previously presented in the paper on the Art of Milling, presented jointly with Carl J. Oxford at the 1921 Annual Meeting of the A.S.M.E., and reiterated the statement made in that paper that the sole criterion of the efficiency with which metal can be removed was the maximum chip thickness for any given material. The spacing of the teeth of the diameter of the cutter had nothing to do with this.

Carl J. Oxford[4] congratulated the authors on presenting scientific facts about an obscure subject. He expressed disappointment, however, because of the narrowness of scope of the research, and the authors' attempt to draw general conclusions from an unquestionably special case, mild steel, one size of milling cutter, and one type of cut only being used. While mild steel could be successfully milled at a speed in excess of 120 ft. per min., high-machineability alloy steels could seldom be milled at a speed higher than 50 ft. per min., although the chip thicknesses of from 0.006 in. to 0.010 in. per tooth might be the same as for mild steel. Productivity of a given size of milling cutter was in direct proportion to the number of teeth in that cutter and the limiting factors for numbers of teeth were:

1 Rigidity of the machine and holding devices

2 Depth or width of cut and the ability of the machine to deliver power to the cutter

3 Structural strength of the cutter.

In discussing Mr. Benzon's paper on the Design of Large Machine Tools, L. R. Meisenhelter[5] pointed out as three important factors in modern large-machine-tool design the importance of the quality and design of gears, the need for centralized control, and the development of proper lubrication.

[1] Mechanical Engineer, New York City. Mem. A.S.M.E.

[2] Consulting Engineer, Philadelphia, Pa. Mem. A.S.M.E.

[3] Prof. Engrg. Mechanics, University of Michigan. Mem. A.S.M.E.

[4] Chief Engineer, National Twist Drill & Tool Co., Detroit, Mich. Assoc-Mem. A.S.M.E.

[5] Mechanical Engineer, Philadelphia, Pa. Assoc-Mem. A.S.M.E.

Engineering and Industrial Standardization

New Plan for Financing Industrial Standardization Is Approved

A NEW plan for financing the industrial standardization work of the United States, which provides for membership dues on the basis of one cent per $1000 of gross receipts, has been formally approved by the Executive Committee of the American Engineering Standards Committee. Twenty of the most influential industrial executives of the country have accepted places on an Advisory Committee which will coöperate with the Ways and Means Committee in the refinancing of the American Engineering Standards Committee.

This report announces a new class of members in the A.E.S.C. to be known as "sustaining members," and provides for them a special service, including information bulletins on developments in standardization work in this country and in every other country where industrial standardization is in progress.

The newly created Advisory Committee of the A.E.S.C. consists of the following men:

W. H. Barr, President, National Founders' Association, Chicago, Ill.
A W. Berresford, General Manager, Cutler-Hammer Company, Milwaukee, Wis.
L. F. Butler, President, Travelers Insurance Company, Hartford, Conn.
Wm. Butterworth, President, Deere and Company, Moline, Ill.
John J. Carty, Vice-President, American Telephone and Telegraph Company, New York, N. Y.
W. W. Coleman, President, Bucyrus Company, South Milwaukee, Wis.
G. B. Cortelyou, President, Consolidated Gas Company, New York, N. Y.
J. K. Cullen, President, Niles-Bement-Pond Company, New York City
J. E. Edgerton, President, National Association of Manufacturers, New York, N. Y.
John R. Freeman, Consulting Engineer, Providence, R. I.
E. M. Herr, President, Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa.
Charles E. Hodges, President, American Mutual Liability Insurance Company, Boston, Mass.
Sidney J. Jennings, President, U. S. Smelting, Refining and Mining Company, New York, N. Y.
J. W. Lieb, Vice-President, New York Edison Company, New York, N. Y.
John B. Morton, President, National Board of Fire Underwriters, Philadelphia, Pa.
Dr. Chas. L. Reese, Chemical Director, E. I. du Pont de Nemours and Company, Wilmington, Del.
E. W. Rice, Jr., Honorary Chairman of the Board, General Electric Company, Schenectady, New York.
Henry D. Sharpe, Treasurer, Brown and Sharpe Manufacturing Company, Providence, R. I.
S. W. Stratton, President, Mass. Institute of Technology, Cambridge, Mass.
Ernest T. Trigg, President, John Lucas and Company, Inc., Philadelphia, Pa.

The American Engineering Standards Committee, which was organized in 1918, has heretofore been financed entirely by dues from the nine technical societies and seventeen national trade associations which with seven departments of the Federal Government constitute its present membership. Annual deficits have been cleared by contributions from individual corporations.

It is expected that the new plan of financing will provide an annual budget of $50,000 for the Standards Committee. As this sum is to be realized from sustaining-membership dues amounting to one-thousandth of one per cent of gross receipts, the total of $50,000 would be spread over industries with aggregate gross annual receipts of five billion dollars. For firms which for any reason prefer to subscribe on the basis of capital rather than gross annual receipts, the recommended basis is one and one-half cents per thousand dollars of aggregate market value of the corporate securities of the firm.

The plan calls for the appointment of an engineer-translator who will provide translations of standards developed in foreign countries for the information service to sustaining members. The new information service will be an elaboration of the work which the A.E.S.C. has been carrying on in a small way, in calling to the attention of coöperating bodies and the technical press the important developments in standardization work, foreign as well as American.

The Ways and Means Committee which drew up this plan of financing consisted of the following members:

Albert W. Whitney, Associate General Manager, National Bureau of Casualty and Surety Underwriters.
A. Cressy Morrison, Vice-President, Compressed Gas Manufacturers Association.
W. W. Nichols, Assistant to the President, Allis-Chalmers Manufacturing Company, and President, Electrical Manufacturers Club.
Frank W. Smith, Vice-President, The United Electric Light & Power Company, and President, National Electric Light Association.

Recent Progress of Some Sectional Committees

Standardization of Shafting. Though this Committee has held no meeting since that of December 5, 1922, it has been by no means inactive. The Sub-Committee headed by Prof. A. H. Beyer has held monthly meetings and is well along toward the completion of Part 2 of its report, namely, the Technique of Shafting Design.

Sub-Committee No. 1, of which Louis W. Williams is chairman, has drafted and circulated a questionnaire to approximately 260 dealers and manufacturers in an effort to ascertain which stock lengths of shafting are preferred. With this questionnaire the manufacturers received another prepared by the American Society for Testing Materials on the present methods employed in manufacturing steel shafting and the minimum values for elastic limit and tensile strength for the different sizes and kind of shafting produced by them. The real purpose of this inquiry was to secure data by which a minimum value of the unit stress for this material might be determined for the use of Professor Beyer's Sub-Committee. The response to these questionnaires has been very gratifying.

Standardization of Plain Limit Gages for General Engineering Work. A letter ballot on the first report to be completed by this Sectional Committee is now being taken. The report covers standard allowances and tolerances for fits in interchangeable manufacture, and is the part of the work assigned to the Sub-Committee of which George T. Trundle is chairman.

The Sub-Committee on Methods of Gaging Manufactured Material, headed by F. O. Hoagland, and the Sub-Committee on Gages and Their Limits, Manufacture and Use, presided over by Earle Buckingham, are both hard at work on their respective sections and promise reports in the near future.

Standardization of Pipe Flanges and Fittings. The all-day session which this Sectional Committee held in December produced many valuable suggestions which have been made good use of in preparing the revised drafts of the four flange standards which are now before the members of the Sectional Committee for approval. These standards are:

1 Cast-Iron Flange and Fittings Standards for 125 lb. Maximum (Saturated) Steam Working Pressure
2 Cast-Iron Flange and Fittings Standards for 250 lb. Maximum (Saturated) Steam Working Pressure
3 Cast-Steel Flange and Fittings Standards for 400 lb. Maximum Steam Working Pressure
  800 lb. Cold-Water Working Pressure—Hydrostatic (No Shock)
  500 lb. Cold-Water Working Pressure—Shock
  800 lb. Air or Gas Working Pressure—Temp. Not Exceeding 100 Deg. Fahr.
4 Cast-Steel Flange and Fittings Standards for 600 lb. Maximum Steam Working Pressure
  1200 lb. Cold-Water Working Pressure—Hydrostatic (No Shock)
  800 lb. Cold-Water Working Pressure—Shock
  1200 lb. Air or Gas Working Pressure—Temp. Not Exceeding 100 Deg. Fahr.

The Sub-Committee on Screwed Fittings is now at work on standard dimensions for cast-steel fittings and is preparing a supplementary report on the Malleable Fittings which will assign tolerances to certain of the dimensions.

Standardization of Bolt, Nut, and Rivet Proportions. At its December meeting this Committee elected as chairman, Arthur A. Norton, assistant professor of mechanical engineering, Harvard

University. It also created an eighth Sub-Committee on Nomenclature which consists of the chairmen of the other seven Sub-Committees, the chairman and secretary of the Sectional Committee and is headed by George S. Case as chairman.

Tentative reports of four of the Sub-Committees are now being considered by the members of the entire Sectional Committee and are available for distribution to those who may desire to look them over. These reports cover:

1 Standards for Large and Small Rivets
2 Standards for Wrench-Head Bolts and Nuts
3 Standards for Slotted-Head Products
4 Standards for Carriage Bolts.

The Sub-Committee on Track Bolts, of which C. W. Squier is now chairman, held a meeting on March 9 and made a very careful survey of the large task assigned to it. A mass of data was laid before the Sub-Committee and this is being carefully studied as the first part of its program.

Safety Code for Mechanical Power-Transmission Apparatus. On February 8 this Sectional Committee held a meeting in New York at which were considered comments which had resulted from the circularization of galley proofs of this safety Code. The correspondence and the discussion at the meeting centered very largely around Rule No. 313 which covers the use of metal belt lacing on belts shifted by hand. A number of representatives of the manufacturers of this material were present and after considerable discussion this rule was revised to read as follows:

> RULE 313—BELT FASTENERS
>
> Belts which of necessity must be shifted by hand and belts within six (6) feet of the floor or working platform which are not guarded in accordance with the intent of this code shall not be fastened with metal in any case nor with any other fastening which by construction or wear will constitute an accident hazard.

The final revision of the text of this Code are now finished and it is before the Sectional Committee for vote. Copies, therefore, will soon be ready for distribution.

Safety Code for Elevators. Sullivan W. Jones, a representative of the A.I.A. on the Sectional Committee and now New York State Architect, was elected chairman of this Committee at the meeting it held on February 19, 1923. At this meeting an Executive Committee was appointed consisting of S. W. Jones, Chairman, O. P. Cummings, Vice-Chairman, J. A. Dickinson, Secretary, M. B. McLauthlin, C. H. Weeks, and K. A. Colahan.

After the Committee had increased its personnel by electing Byron Cummings and D. L. Lindquist as members at large, it turned its attention to the report of its Plan and Scope Committee, of which Bassett Jones was chairman. As the result of the discussion which followed the presentation of this report, the chairman was directed to appoint two Sub-Committees and to develop with the assistance of the Executive Committee a strong Advisory Committee. The first of these Sub-Committees is to be charged with the interpretation and revision of the Code. It will maintain contact with the various regulatory and other interested bodies for the purpose of answering specific questions about the Code, to issue special interpretations and to recommend such changes to Sub-Committee No. 2 which have been suggested by these regulatory bodies or others.

The task assigned to Sub-Committee No. 2 is fourfold; (*a*) to indicate the sections of the present A.S.M.E. Code which apply to existing installations; (*b*) to revise the present Code wherever necessary to cover completely new installations; (*c*) to draft a set of rules for operation and inspection; and (*d*) to compare existing rules of various states and cities so that the Sectional Committee shall at all times be thoroughly informed.

The Federated American Engineering Societies

Labor-Saving Machinery

AT A meeting of the Committee on Procedure of the American Engineering Council, held on February 16, 1923, the advisability of making an intensive study of labor-saving devices was brought up for consideration. Walter S. Moody, of the General Electric Company, Pittsfield, Mass., who, in a communication addressed to President Cooley, first suggested such an investigation as a means of overcoming the present labor shortage in the United States, stated that in his opinion a large amount of labor now performed by hand could be accomplished by power-driven devices, thereby releasing great numbers of men for other and higher-grade work. He advocated a general survey of industry for the purpose of selecting kinds of work now done manually where machinery might be employed, followed by the coöperation of industries and suitable engineering organizations in the development of the necessary devices.

The Committee believed the matter to be of very great importance and after discussion voted that it be referred to The American Society of Mechanical Engineers to ascertain, first, if the A.S.M.E. considers such an investigation opportune and practical; and second, if that society will undertake the study or, in case it does not desire to do so, if it advises that the work be undertaken by American Engineering Council, and in what form.

Reforestation

REFORESTATION, involving such agricultural and industrial problems as soil conservation, flood control, and power development, is one of the projects now being studied by a special F.A.E.S. committee, in coöperation with the Government. The chairman of this committee, which was appointed to assist in developing a comprehensive constructive plan of reforestation, is Charles H. McDowell of Chicago; other members are S. H. McCrory, chief of the Division of Agricultural Engineering, U. S. Department of Agriculture. W. H. Hoyt, of Duluth, Minn., and J. C. Ralston, of Spokane, Wash.

Chief McCrory, in an announcement of the project issued recently through Dean M. E. Cooley, president of the Federation, stated that in the past land development and utilization has depended largely upon local initiative but that consideration must soon be given to the probable needs of the United States for crop, pasture, and forest lands. America's supply of timber, which has been decades in growing, is rapidly being exhausted. If timber is to be grown to replenish the supply, a considerable acreage must be permanently devoted to this purpose. He pointed out that the easily developed tillable land has nearly all been brought into use, and emphasized the need for the most effective utilization of undeveloped lands, clearing, draining or irrigating for crop and pasture land, or reserving for timber growth.

The committee believes it essential that mountainous regions such as the Appalachian area be kept timbered in order to minimize erosion. The watersheds of rivers must not be divested of timber, otherwise erosion will be rapid and power development retarded. The committee is supporting legislation for the establishment of a national hydraulic laboratory to study flood control.

Eyesight Conservation Council

A CONTINUATION of one phase of the work of the Committee on the Elimination of Waste in Industry is being conducted by the Eye Sight Conservation Council of America, which is directing a campaign to eliminate economic and physical losses due to poor eyesight in schools and factories, and to conserve vision. The Waste Report showed that poor eyesight among workers causes heavy annual economic losses. A similar condition exists in schools.

L. W. Wallace, executive secretary of the F.A.E.S. was reëlected president of the Eye Sight Conservation Council at its annual meeting, held in New York, February 5, and Guy A. Henry, of New York, was reëlected general director. Others recently appointed to the governing bodies of the Council are Secretary Davis, of the U. S. Department of Labor, and Prof. F. C. Caldwell of the Department of Electrical Engineering, Ohio State University. Among those serving on the Board of Directors and the Board of

Councilors are Prof. J. W. Roe, New York University, president of the Society of Industrial Engineers; Dr. Morton G. Lloyd, chief of the Safety Section of the U. S. Bureau of Standards and vice-president of the American Society of Safety Engineers; and G. E. Sanford, of West Lynn, Mass., past-president of the American Society of Safety Engineers.

The personnel of those assisting in the movement includes also representatives of the U. S. Department of Education, the U. S. Public Health Service, and numerous other engineering and educational bodies.

Passage of Bill for Reclassification of Governmental Positions and Salaries

ONE OF THE closing acts of the sixty-seventh Congress was the passage of the Sterling-Lehlbach Bill for the Reclassification of Governmental Positions and Salaries, H. R. 8928. As a means of maintaining a high standard of scientific and technical service of the Government, this bill has received the active support of the F.A.E.S. through its committees on patents and on reclassification and compensation of engineers. A delegation of engineers headed by Col. J. H. Finney, a member of the Patents Committee, was instrumental in securing its passage before Congress adjourned.

Although the law will not go into effect until July 1, 1924, it undoubtedly has an immediate benefit. High-grade professional men already in Government employ, with the definite salary increase before them, will feel that they can "stay by the ship," and others of similar type will be drawn into Government service, filling many vacancies which low salaries have caused. Until the law goes into effect the present basic rates of pay, plus the $240 bonus that has been paid for the last six years, will continue.

The bill as finally passed by the House carries revisions made by the Senate. In the Patent Office, for instance, primary examiners will receive $5040 instead of $4600. The enactment of the Lambert Patent Office Bill, early in 1922, prevented the complete collapse of the Patent Office; the salaries under the Sterling-Lehlbach law will attract for new appointments a high grade of men. In the course of time the whole examining corps will be raised in quality. The law will equally benefit all other professional and scientific branches of the Government.

Engineering organizations have done much to secure the passage of these two important bills. Engineers may be justly proud of their part in completing the task set before them. Such success should make them realize that they do have a strong national influence and should go a long way toward bringing about whatever seems to them just and wise.

Reorganization of Government Departments

THE OUTLINE of the plan for the reorganization of the executive departments of the U. S. Government as recommended by the President and the Cabinet at the request of the Joint Committee on Reorganization was recently placed in the hands of that committee for consideration. The outstanding recommendations are as follows:

I The coördination of the Military and Naval Establishments under a single Cabinet officer, as the Department of National Defense.

II The transfer of all non-military functions from the War and Navy Departments to civilian departments—chiefly Interior and Commerce.

III The elimination of all nonfiscal functions from the Treasury Department.

IV The establishment of one new department—the Department of Education and Welfare.

V The change of the name of the Post Office Department to Department of Communications.

VI The attachment to the several departments of all independent establishments except those which perform quasi-judicial functions or act as service agencies for all departments.

The suggested changes of particular interest to engineers are found in the Department of the Interior and Commerce. The Interior Department is given two major functions: the administration of the public domain and the construction and maintenance of public works. The subdivisions of the department are grouped accordingly under two assistant secretaries. The Bureau of Mines and the Patent Office are transferred to the Department of Commerce, and the non-military engineering activities of the War Department, the Supervising Architect's Office of the Treasury Department, the Bureau of Public Roads of the Department of Agriculture, and the Federal Power Commission, an independent establishment, are all transferred to the Department of the Interior.

The three major functions given to the Department of Commerce are the promotion of industry, the promotion of trade, and the development, regulation, and protection of the merchant marine, each division under an assistant secretary. In addition to the transfer of the Bureau of Mines and the Patent Office from the Department of the Interior, the Department of Commerce is given the Lake Survey, the Inland and Coastwise Waterways Service, the supervision of New York Harbor, the Hydrographic Office and the Naval Observatory, and the Life Saving Service.

While this plan does not provide for the establishment of a National Department of Public Works, which has been advocated by engineering organizations, the proposed transfers have in view the same object, that of bringing together related activities. The outline has been studied in detail by the Federation's Committee on Government Reorganization as it Relates to Engineering Matters, of which J. Parke Channing, New York, is chairman, and was reported upon to the Executive Board, at its meeting in Cincinnati, March 23–24, 1923.

Discussion on Effect of Pulsations on Flow of Gases

(*Continued from page 229*)

perhaps nearly eliminated, by the use of the proper amount of throttling observing at the same time that equal spaces were provided at the manometer connections for varying quantities of flow. They concluded that it was an extremely doubtful and uncertain method, if not wholly dangerous, to rely too fully on such expedients for reducing the pulsating error.

The question raised in regard to the relative effect of the pulsation on the static pressure was answered, the authors believed, in the reply to Mr. Packard.

The authors agreed with Mr. Hodgson and Mr. Packard that the velocity of propagation of the pulsating wave approached that of sound in the flowing fluid plus the velocity of the flowing air. However, in their opinion, as based on their experiments they were not willing to concede that the pressure pulsation had a less effect than the velocity pulsation in meter installations where these pulsations acted on manometers with static connections at right angles and even with the inside of the pipe. From the "dead-end" flow experiments it would also seem evident that the pressure pulsation effect was many times greater than that due to the velocity pulsation.

The statement was made by Mr. Hodgson that the simplest way of reducing the pulsation by the use of a capacity combined with throttling was not mentioned by the authors. In conclusion they had stated that the "muffler" device, a combination of capacity, or volume, with throttling was probably "the most effective device for the mechanical elimination of pulsations." The capacity or "muffler" was always inserted between the disturbing element and the meter. It would not seem advisable in their opinion, to insert the throttling device in the line below the meter. It would seem better to eliminate the pulsations as far as possible before the meter was reached.

The authors regretted that Mr. Hodgson's paper containing his latest investigations involving the development of a formula for pulsating flow had not been available for examination and study until after their paper was written. Mr. Hodgson was to be congratulated in being able to reduce the results of his researches to a working formula which showed the factors involved in their proper relation. They were in full accord with his opinion that the only sure way to meter pulsating flow was to reduce the pulsations by means of suitable capacity and throttling, and they likewise believed that even the formula proposed, or any similar formula, while serving in a general way could not be too rigidly applied in practice. Each installation would present its own peculiar problem.

Meetings of Other Societies

American Institute of Mining and Metallurgical Engineers

The 127th meeting of the American Institute of Mining and Metallurgical Engineers was held in New York, February 19–22, 1923. So valuable was each of its sessions to the group of engineers interested in the particular problem with which it dealt, that no session can be called outstanding.

The opening technical session on Monday morning was on the subject of ground movement and subsidence, which has received comparatively little treatment during recent years. Much interest was evidenced in the papers presented and the session was continued in the afternoon.

A meeting for the organization of a Division on Petroleum and Gas was held on the morning of the first day of the convention and three entire sessions were devoted to symposiums on this subject. Twenty-three papers covered oil developments throughout the world during 1922, and six papers dealt with miscellaneous oil-development problems.

Based upon replies to questionnaires sent out to a large number of mining companies during 1922, a number of papers on mining methods were presented at four mining sessions, one of which was devoted entirely to coal.

Two iron and steel sessions were held. At the first the A.S.T.M. Tentative Specifications for Foundry Pig Iron were presented and the discussion led by Dr. Richard Moldenke, Watchung, N. J. Dr. Moldenke is chairman of the A.S.T.M. Committee on Cast Iron, at whose request the specifications were presented at the Institute meeting. The specifications were thought to be liberal in character and were favorably received.

At the same session Norman B. Pilling, Research Department, Westinghouse Elec. & Mfg. Co., gave an illustrated talk on Low-Temperature Brittleness in Silicon Steels. He stated that temporary ductility may be obtained by carrying on cutting or deformational operations at temperatures slightly above atmospheric, the temperature depending on the steel composition, and that brittleness is only slightly modified by heat treatment.

Speaking at the second session on iron and steel, W. R. Bean, Naugatuck, Conn., stated that the deterioration of malleable in the hot-dip galvanizing process is intimately connected with the phosphorus and silicon contents of the iron. In general, low-phosphorus (under 0.15 per cent), low-silicon (under 0.80 per cent) iron will withstand the process best and show practically no deterioration. High-phosphorus, high-silicon irons are practically certain to deteriorate and be embrittled by galvanizing. A second paper at this session, dealing with the influence of temperature, time, and rate of cooling on physical properties of carbon steel, gave the results of an investigation of the heat treatment of carbon steels carried out under the auspices of the Committee on this subject of the Engineering Division of the National Research Council.

There was also a joint session on iron and steel and coal and coke, at which blast-furnace cokes and coke ovens were considered.

Joint sessions were held with the Industrial Relations Committee and Mining Section of the National Safety Council, at which hoisting ropes and mine-fire prevention were the main subjects considered. An interesting paper on Non-Destructive Testing of Steel Hoisting Rope by Raymond L. Sanford, physicist of the National Bureau of Standards, was presented and discussed. It outlined work soon to be undertaken by that bureau to develop if possible suitable magnetic methods of testing steel hoisting ropes.

Technical education was discussed at a joint meeting with the Mining and Metallurgical Society of America. One of the papers was by E. P. Mathewson, consulting metallurgist of New York and incoming president of the Institute, who gave his opinions on training engineering students, based upon his contact during thirty-five years with graduates of engineering schools all over the world.

Mechanical engineers will be chiefly interested in the papers presented under the auspices of the Institute of Metals Division. Dr. Walter Rosenhain, head of the metallurgical department of the National Physical Laboratory at Teddington, England, delivered the second annual Institute of Metals lecture, on the subject of Solid Solutions, to a large audience on February 19. The high quality of Dr. Rosenhain's lectures has been appreciated by many in this country who have heard him during recent months at various technical and educational institutions. It was announced that Dr. Zay Jeffries will deliver the lecture next year.

The Nature of Solid Solutions was the title of a paper by Edgar C. Bain, of the General Electric Company, Cleveland, Ohio. E. H. Dix, Jr., described methods of polishing aluminum and its alloys for metallographic study as employed in the metallurgical laboratory of the Engineering Division of the U. S. Air Service, McCook Field, where he is chief of the Metals Branch. Junius D. Edwards, assistant director of research, Aluminum Company of America, spoke on the thermal properties of aluminum-silicon alloys, dealing with the accurate determination of densities of aluminum alloys containing variable amounts of silicon and presenting data on crystallization shrinkage, total solid shrinkage, and the tendency to form pipe.

Christopher H. Bierbaum, vice-president of the Lumen Bearing Company, Buffalo, N. Y., in an address on bearing metals, described an instrument developed for use in determining the relative hardness of individual crystals in the bearing metal and the journal.

Among the other papers was one on tests on high-tin bearing metals, including remarks on the chemical composition and microstructure of suitable alloys, directions for casting in bronze shells, and data on tests of commercial alloys for hardness, compressive strength, and coefficient of friction; a paper on the thermal conductivity of some industrial alloys; and one on the bright annealing of copper wire.

At the annual dinner of the Institute, at which Prince Gelasio Gaetani, the Italian ambassador, was guest of honor, the James Douglas medal was awarded to Frederick Laist, manager of the reduction plant of the Anaconda Copper Mining Co., Butte, Mont., for achievements in non-ferrous metallurgy.

E. P. Mathewson was elected president of the Institute to succeed Col. Arthur S. Dwight.

American Institute of Electrical Engineers

Out of the unusually large number of technical papers presented at the Eleventh Midwinter Convention of the American Institute of Electrical Engineers, held in New York, February 14–17, 1923, are several which are of interest to mechanical engineers. E. J. Blake, electrical engineer for Gould Coupler Co., Depew, N. Y., reviewed the circumstances which have made train control an acknowledged problem and stated the characteristics desired in automatic train-control devices. Automatic control, he believed, should act as a check on manual control, not as a substitute. It should conform to established safe signaling practices, should be designed for the severe conditions of railway operation, should not conflict with existing signals or otherwise introduce new hazards, should so far as possible conform to existing clearance lines and should not impede traffic. Methods of transmitting and indication of track conditions to the train which he described were (1) by intermittent mechanical and electric contact; (2) by intermittent induction through the use of permanent or electromagnets; and (3) by continuous induction from the rails. Relations between the type of controlling action and traffic capacity were also discussed by Mr. Blake.

Applications and Limitations of Thermocouples for Measuring Temperatures was the subject of an address by Irving B. Smith, Research Department, Leeds & Northrup Co., Philadelphia, Pa. He classified temperature measurements under the heads of feed-water, boiler water, economizer, flue gas, superheated steam, bearings, generator windings, transformer windings, and cables. The sources of error that may arise in making temperature measurements with the thermocouple were outlined as residing in thermocouple calibration, instrument calibration, thermocouple circuit, radiation losses, conduction losses, parasitic e.m.f., temperature lag, cold-junction temperature, measured temperature variable, and measured temperature not representative. Mr. Smith considered these classifications and sources of error in detail.

At the same session J. W. Legg, consulting engineer for Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., presented a paper

on the Expansion of Oscillography by Portable Instrument, describing fully the oscillograph in its redesigned form and discussing its possibilities. The new instrument has been constructed along the same general principles as the early portable oscillograph, but has been reduced in bulk about fifty per cent and considerably improved. It is complete in one unit and is designed to operate from any standard 6-volt storage battery.

A feature of the meeting which aroused great interest was the joint session with Chicago on the evening of February 14, made possible by two-way loud-speaking telephone installations. The proceedings of the meeting were also broadcast. Speakers on the use of public-address systems with telephone lines discussed the various applications of this novel combination, stated requirements for the lines, showed the circuit arrangements used, and described some of the important operating features. The application of the public-address-system apparatus and methods to radio broadcasting were also briefly discussed.

Inspection trips to the McGraw-Hill Company, the Pennsylvania Exchange of the New York Telephone Company, the Bell System Research Laboratories, and the A.T. & T. Broadcasting Station were made during the convention.

American Society of Heating and Ventilating Engineers

Following a business session in New York on January 23, the American Society of Heating and Ventilating Engineers held the first technical session of its twenty-ninth annual meeting at the Bureau of Standards in Washington, January 24. The morning was devoted to Bureau of Standards papers on subjects of great importance to heating and ventilating engineers, which were based on current research in the laboratories of the Bureau; the afternoon was given over to an inspection of the laboratories and an explanation of the methods of investigation. The papers presented were Heat Transmission of Building Structures, by M. S. Van Dusen; The Testing of Anemometers, by O. J. Hodge; and Tests of Radiator Return Line Valves, by W. F. Stutz. S. H. Ingberg gave a talk on Fire Tests of Structural Materials.

On January 25 a research session was held at the Bureau of Standards at which F. Paul Anderson spoke on The Research Laboratory, and papers on the physiological reactions of humans to high temperatures and high humidities, equal-comfort lines for still-air conditions, the capacities of steam-heating risers as affected by critical velocities of steam and condensate mixtures, and the Anderson and Armspach dust determinator were presented. The latter paper included information on results obtained in a series of tests upon the dust determinator, as well as a description of the apparatus.

The other sessions of the meeting were held at the Hotel Washington on January 24 and 26. At the first, M. S. Cooley of the Bureau of Yards and Docks, told how the new gun shop at the U. S. Navy Yard is heated, describing in detail the hot-blast system which is provided with fans and heaters in roof spaces and distributing ducts alongside of columns. C. R. Denmark, engineer at the Smithsonian Institution, gave details of the forced hot-water circulation and air-exhaust systems in the Natural History Building, of the U. S. National Museum. Nelson S. Thompson, chief mechanical and electrical engineer, Office of Supervising Architect, U. S. Treasury Department, pointed out the simplicity of the heating and ventilating systems installed at the Bureau of Engraving and Printing.

Heating and ventilating sessions were held on the closing day of the convention, at which a number of interesting papers and reports were heard. F. B. Rowley, professor of mechanical engineering at the University of Minnesota, gave a report of tests of five distinct types of roof ventilators and described the method and apparatus used in making the tests. Dr. George T. Palmer, of the Michigan Department of Health, outlined ventilation practice since 1850 and indicated what developments may be expected in the near future. H. L. Dryden discussed wind-tunnel tests of roof ventilators.

The program included various sight-seeing and inspection trips to the many beautiful institutions which are to be found both in and around Washington.

LIBRARY NOTES AND BOOK REVIEWS

Analytic Geometry. By Clyde E. Love. Macmillan Co., New York, 1923. Cloth, 5 × 8 in., 306 pp., $2.25.

A textbook for beginners, covering the elements of plane and solid analytic geometry. The emphasis is placed on geometry rather than on analysis to a greater extent than usual, and the course is intended to give the student, first, a knowledge of simple fundamental methods, and then to extend, adapt, and generalize these as need arises.

Comparison of British and American Foundry Practice. By P. G. H. Boswell. University Press of Liverpool, Liverpool; Hodder & Stoughton, London, 1922. Paper, 6 × 9 in., 106 pp., illus., diagrams, tables, 4s. 6d.

The author of this work was sent to the United States in 1918 by the British Ministry of Munitions of War, to investigate American foundry practice, particularly as regards the use of sands for steel molding. For this purpose he visited the principal iron, steel, and brass foundries, the quarries and other sources of refractories, and research laboratories. His report discusses the casting of metals, desiderata in molding sands, the molding-sands used in America, bonding of molding sands and silica sands for furnace hearths. British and American methods are compared. Tables give chemical and mechanical analyses and the mineralogical composition of a number of American and European sands.

Construction and Exploitation des Grandes Réseaux de Transport D'Energie Électrique a Très Haute Tension. Proceedings of the International Conference held in Paris November 21–26, 1921. L'Union des Syndicate de L'Electricite, Paris, 1922. Cloth, 7 × 10 in., 1176 pp., illus., diagrams, 100 fr.

The reports of this conference are now available in a well-printed volume of nearly twelve hundred pages containing a review of its organization and purpose, a general report of its activities, and the text of the sixty-eight reports presented before it. These reports include summaries of the legislation pertaining to high-tension transmission in various countries, descriptions of existing systems and projects, papers upon various topics connected with current production and transformation, the construction of high-tension lines, and the exploitation, protection, and safeguarding of transmission systems. The conference was attended by delegates from twelve countries. The papers and discussions give a wide survey of the present status of high-tension transmission.

Descriptive Geometry. By Lawrence E. Cutter. First edition. McGraw-Hill Book Co., London and New York, 1923. Cloth, 6 × 9 in., 244 pp., diagrams, $2.50.

In descriptive geometry there are two general methods of applying its fundamental principles: one, the method of rotation; the other, the method of choosing new projection planes. The present book is the first, the author states, prepared on the latter plan, which he considers superior in utility to the first plan, and equally good in point of mental discipline.

In it theory has been reduced to four simple principles, covering one page of the text. The use of these is illustrated through the detailed solution of problems from the fields of civil, mechanical, and mining engineering, and of architecture. Original exercises for solution are given.

Diesel Engine. By A. Orton. Isaac Pitman & Sons, London and New York, 1923. (Pitman's Technical Primers.) Cloth, 4 × 6 in., 111 pp., diagrams, tables, $0.85.

The aim of this book is to act as an introduction for those intending to study the subject thoroughly, and also to serve as a broad but fairly complete treatment for those who seek just sufficient

knowledge to understand and apprehend the principles of working, construction, and operation. The author has endeavored to treat the subject in the simplest possible manner, without omitting anything of vital importance. A bibliography, confined to British publications, is included.

OPTICAL METHODS IN CONTROL AND RESEARCH LABORATORIES. By J. N. Goldsmith and others. Vol. 1, Second edition. Adam Hilger, London, 1923. Limp cloth, 6 × 10 in., 56 pp. illus., 1s 6d.

The optical methods here dealt with are those employing spectroscopes, spectrophotometers, refractometers and polarimeters. The book is intended to provide the works chemist with a guide to the selection of these instruments, an introduction to their use in metallurgy and analytical chemistry and an index to sources of further information concerning their use in research and control laboratories. The book is confined to these indications of the usefulness of these instruments, and does not include detailed descriptions of their design or their techniques; for information on these points, references are given to other publications.

PRINCIPLES AND PRACTICE OF TOOTHED GEAR WHEEL CUTTING. By George W. Burley. Scott, Greenwood & Son, London; D. Van Nostrand Co., New York, 1922. Cloth, 6 × 9 in., 460 pp., illus., diagrams, tables, $8.

This book deals with the fundamental principles of all the several descriptions of toothed-wheel gearing now in use; the measurement of toothed-wheel gears, generating and non-generating methods for forming gear-wheel teeth mechanically, and the machines and tools that apply these methods. An attempt is made to treat both the theoretical and practical side of the subject.

The book is intended for students of engineering, apprentices, and machinists. The subject is not dealt with from the viewpoint of the designer.

SCIENTIFIC MANAGEMENT. By Horace Bookwalter Drury. Third edition. Columbia University, New York, 1922. (Studies in history, economics and public law). Cloth, 6 × 9 in., 271 pp., $2.75.

Dr. Drury's account of the development of scientific management appeared originally in 1915. In 1918 a revised edition appeared which corrected the errors of the original edition and extended the account down to the latter date. The present edition is substantially a reprint of the 1918 edition, with a long introduction in which later tendencies and the present situation are described.

The volume is divided into two sections, historical and critical. The historical section gives an account of the genesis of scientific management, of the leaders in its development and of the trades and plants that adopted it. The critical section discusses its effect on productivity, on the labor problem and on the worker.

STEAM POWER. By C. F. Hirshfeld and T. C. Ulbricht. Second edition. John Wiley & Sons, New York; Chapman & Hall, London, 1922. Cloth, 5 × 8 in., 474 pp., illus., diagrams, $3.25.

In preparing this textbook the authors have attempted to collect in a comparatively small book such parts of the field of steam power as should be familiar to engineers whose work does not require that they be conversant with the more complicated thermodynamic principles considered in advanced treatises. They have therefore eliminated mathematical treatment as far as possible, and confined themselves to giving a correct viewpoint with regard to the use of heat in the power plant, to supplying what is required to solve common power-plant problems and to describing the more common types of apparatus.

TREATISE ON THE THEORY OF BESSEL FUNCTIONS. By G. N. Watson. University Press, Cambridge, 1922. Cloth, 7 × 11 in., 804 pp., tables, $16.

The author states that this book has been designed with two objects in view. The first is the development of applications of the fundamental processes of the theory of functions of complex variables, for which purpose Bessel functions are admirably adapted. The second is the compilation of a collection of results which would be of value to the increasing number of mathematicians and physicists who encounter Bessel functions in the course of their researches. Such a collection seems to be demanded by the greater abstruseness of properties of Bessel functions which have been required in recent years in various problems of mathematical physics.

While the endeavor has been made to give an account of the theory of Bessel functions which a pure mathematician would regard as fairly complete, the author consequently has also endeavored to include all formulas which, although without theoretical interest, are likely to be required in practical applications. A very full bibliography is included.

TWELVE-HOUR SHIFT IN INDUSTRY. By Federated American Engineering Societies. Committee on Work-Periods in Continuous Industry. E. P. Dutton & Co., New York, 1922. Cloth, 6 × 8 in., 302 pp., tables, $3.50.

The investigations reported upon in this volume were undertaken by the Federated American Engineering Societies in 1921. The objects were to ascertain the extent of two-shift work in continuous-process industries other than the manufacture of iron and steel, the experience of manufacturers who had changed from two-shift operation to some other system, and to study the technical aspects of changing from the two-shift to the three-shift system in the iron and steel industry.

This volume includes a report on the first two points prepared by Dr. Horace B. Drury and one on the third point by Mr. Bradley Stoughton, as well as a brief general summary of the conclusion to be drawn from their studies. These are favorable to the two-shift system.

VECTOR CALCULUS, WITH APPLICATIONS TO PHYSICS. By James Byrnie Shaw. D. Van Nostrand Co., New York. Cloth, 5 × 8 in., 314 pp. $3.50.

Embodies the author's lectures to graduate students. The attempt has been to give a text to the mathematical student on the one hand, in which every physical term beyond mere elementary terms is carefully defined. On the other hand, for the physical student there is a large collection of examples and exercises which will show him the utility of the mathematical methods. The system adopted is algebraic.

New Revisions of A.S.M.E. Boiler Code, 1923

(Continued from page 263)

Part II

The following action was taken on this section:

Voted: That Part II, on Existing Installations, be placed in the Appendix as suggested rules for the old boilers.

Revisions on Locomotive Boiler Code

L-53 REVISED:

L-53 ALL HOLES IN BRACES, LUGS AND SHEETS FOR RIVETS OR STAYBOLTS SHALL BE DRILLED, OR THEY MAY BE PUNCHED, AT LEAST $^1/_8$ IN. LESS THAN FULL DIAMETER FOR MATERIAL NOT MORE THAN $^5/_{16}$ IN. THICK, AND AT LEAST $^1/_4$ IN. LESS THAN FULL DIAMETER FOR MATERIAL MORE THAN $^5/_{16}$ IN. THICK.

SUCH HOLES SHALL NOT BE PUNCHED IN MATERIAL MORE THAN $^3/_8$ IN. THICK.

FOR FINISHING THE RIVET HOLES, THE PLATES, BUTT STRAPS, BRACES, HEADS AND LUGS SHALL BE FIRMLY BOLTED IN POSITION BY TACK BOLTS, FOR FINAL DRILLING OR REAMING TO FULL DIAMETER.

THE FINISHED HOLES MUST BE TRUE, CLEAN AND CONCENTRIC.

L-62 REVISED:

L-62 Each safety valve shall be plainly marked by the manufacturer IN SUCH A WAY THAT THE MARKINGS WILL NOT BE OBLITERATED IN SERVICE. The markings may be stamped on the CASING [body], cast on the CASING [body], or stamped or cast on a plate or plates permanently secured to the CASING (body), and shall contain the following:

a The name or identifying trademark of the manufacturer
b The nominal diameter
c The steam pressure at which it is set to blow
[*d* Blow down, or difference between the opening and closing pressures]
d [*e*] The weight of steam discharged in pounds per hour at a pressure 3 per cent higher than that for which the valve is set to blow
e [*f*] A.S.M.E. Std.

THE ENGINEERING INDEX

(Registered United States, Great Britain and Canada)

Exigencies of publication make it necessary to put the main body of The Engineering Index (p. 111-EI of the advertising section) into type considerably in advance of the date of issue of "Mechanical Engineering." To bring this service more nearly up to date is the purpose of this supplementary page of items covering the more important articles appearing in journals received up to the third day prior to going to press.

AIRPLANES

Bristol. The Bristol 3-Seater Airplane. Aerial Age, vol. 16, no. 3, Mar. 1923, pp. 140–141, 2 figs. Taxiplane designed to compete economically with road transport. See also Aviation, vol. 14, no. 10, Mar. 5, 1923, p. 273, 2 figs.

Sailplanes. The Development of the Hannover Sailplane, Georg H. Madelung. Soc. Automotive Engrs.—Jl., vol. 12, no. 1, Jan. 1923, pp. 77–85, 15 figs. Difference between airplane-racer and sailplane problems; features of early gliders; analysis of rate of descent; section and best-chord determination; why it was not advisable to build biplane; form and arrangement of body; model test; constructional details; torque; static tests and safety factor.

ALLOY STEELS

Heat Treatment. The Heat Treatment of Alloy Steels, R. R. Moore and E. V. Schaal. Forging & Heat Treating, vol. 9, no. 2, Feb. 1923, pp. 113–121, 40 figs. Effect of heat treatment upon metallographic and physical characteristics of chrome-nickel, chrome-vanadium and chrome-molybdenum steels; results obtained at various drawing temperatures.

AVIATION

Experimental Work, Bureau of Standards. Aviation Work of the Bureau of Standards, Fay C. Brown. Aerial Age, vol. 16, no. 3, Mar. 1923, pp. 115–119, 5 figs. Work is for most part in coöperation with Army & Navy Air Service and Nat. Advisory Committee for Aeronautics; deals with aircraft structure and instruments, power plants and radiators; the aeronautical safety code.

BEARINGS, ROLLER

Spherical. Advantages of the Spherical Type Roller Bearing, H. E. Brunner. Ry. Age, vol. 74, no. 9, Mar. 3, 1923, pp. 517–519, 3 figs. New design which combines self-alignment, low frictional resistance and high capacity.

BOILER FEEDWATER

De-Aeration, Closed-Feed. A New Closed Feed System. Engineer, vol. 135, no. 3502, Feb. 9, 1923, pp. 155–156, 5 figs. System developed by Hick, Hargreaves & Co., consists of feed piping and auxiliaries which are arranged so that only de-aerated water is supplied to economizers and boilers.

BOILER FURNACES

Design for Drying Fuel. Furnace Design for Effective Drying, Zuce Kogan. Power Plant Eng., vol. 27, no. 5, Mar. 1, 1923, pp. 265–268, 8 figs. Special arrangements of grates and arches are essential for drying fuel with high-moisture content in boiler furnace.

BOILERS

Atmos. Superpressure. Swedish Boiler Operates at Pressure of 1500 Pounds, Edvin Lundgren. Power, vol. 57, no. 7, Feb. 13, 1923, pp. 238–241, 5 figs. Among features of Atmos. boiler, designed by J. V. Blomquist, are steam generation from centrifugally formed shells of water in rotating tubes of 12 in. diam. and evaporation of 60 lb. of water per hr. per sq. ft. of steam-making surface.

DIE CASTINGS

Metals for. Metals Used for Die-castings, A. G. Carman. Machy. (N. Y.), vol. 29, no. 7, Mar. 1923, pp. 516–518, 1 fig. Babbitts, zinc-base metals and aluminum alloys for die castings; specifications giving limits of weight and accuracy for die castings made from different metals.

DIES

Built-up. Advantages of Built-up Die Construction, C. E. Stevens. Machy. (N. Y.), vol. 29, no. 7, Mar. 1923, pp. 528–530, 5 figs. Points to be considered in designing dies, which have been incorporated in set of dies for producing rheostat base herein described.

EVAPORATION

Compression, Method of. Evaporation by Compression, Wilhelm Gensecke. Chem. & Met. Eng., vol. 28, no. 10, Mar. 7, 1923, pp. 448–456, 15 figs. Process now in operation in several important industrial plants in Europe, where it has proved a valuable factor in heat economy; also close study of heat consumption usually calculated in sugar industry.

FLOW OF FLUIDS

Thin Slits. Leakage Through Thin Clearance Spaces, Edgar Buckingham. Engineering, vol. 115, no. 2982, Feb. 23, 1923, pp. 225–227, 1 fig. Results of experiments on slits of rectangular section by Frank F. Fergusson; notes on stream-line flow; change to turbulent flow; comparison of rectangular with annular slits; equations for computing resistance of thin, smooth, annular channels. Published by permission of U. S. Bur. of Standards.

FORGING

Flow of Metal During. The Flow of Metal During Forging, Harold F. Massey. Forging & Heat Treating, vol. 9, nos. 1 and 2, Jan. and Feb. 1923, pp. 25–30 and 122–127, 33 figs. Discussion of flow of metal when forged in hot state; special emphasis given to relation existing between action of forging by press and hammer. Paper read before Manchester Assn. Engrs.

GRINDING

Centerless. The Production of Small Parts by Centerless Grinding, Howard Campbell. Am. Mach., vol. 58, no. 10, Mar. 8, 1923, pp. 357–359, 10 figs. Examples of centerless grinding, together with figures on production; methods of handling odd work; grinding tungsten bars.

INDUSTRIAL MANAGEMENT

Engineering Department. The Successful Operation of an Engineering Department, W. E. Irish. Indus. Management (N. Y.), vol. 65, no. 3, Mar. 1923, pp. 136–141. Discussion of actual operation of department organized on principle described in previous articles.

Production Planning. How Production Planning Cuts Costs, H. S. Owen. Indus. Management (N. Y.), vol. 65, no. 3, Mar. 1923, pp. 182–187, 13 figs. Describes system of planning which, in actual operation, has produced very profitable results.

INTERNAL-COMBUSTION ENGINES

Explosion Tests. Internal Combustion Heat Losses and Specific Heat of Working Fluid, Wm. J. Walker. Engineer, vol. 135, no. 3504, Feb. 23, 1923, pp. 191–192, 1 fig. Discusses extent of transparency of combustion products of normal mixture to its own radiated heat; test determinations of values of specific heats at constant pressure and volume.

IRON CASTINGS

Annealing. Effect of Annealing Gray Iron, J. F. Harper and P. S. MacPherran. Foundry, vol. 51, no. 5, Mar. 1, 1923, pp. 176–180, 11 figs. Test bars heated at different temperatures for various lengths of time show changes in strength and hardness caused by annealing; results charted and micrographs shown. Paper before Am. Foundrymen's Assn.

Softening Gray Iron by Annealing, E. Piwowarsky. Forging & Heat Treating, vol. 9, no. 2, Feb. 1923, pp. 127–129, 1 fig. Object and theory of annealing of cast iron; experiments to soften by annealing; most certain method to accomplish this end. Translated from Stahl u. Eisen, Sept. 28, 1922.

LABORATORIES

Hydraulic Testing. A Well-Designed Hydraulic Testing Laboratory, John S. Carpenter. Power, vol. 57, no. 9, Feb. 27, 1923, pp. 331–332, 5 figs. Describes public testing flume built by Holyoke Water Power Co. at Holyoke, Mass.; laboratory equipment.

LOCOMOTIVES

Mikado. Most Powerful Mikados on the Lackawanna Railroad. Ry. Rev., vol. 72, no. 7, Feb. 17, 1923, pp. 279–285, 10 figs. partly on supp. plate. Latest Mikado-type locomotives develop 79,200 lb. tractive effort. See also Ry. Age, vol. 74, no. 9, Mar. 3, 1923, pp. 510–513, 2 figs.

Pacific and Mikado. Locomotives for Brazilian Centennial Exhibition. Ry. Age, vol. 74, no. 8, Feb. 24, 1923, pp. 467–468, 3 figs. Two Pacifics and a Mikado displayed by Am. Locomotive Co.

Superheater. Southern Railway—Three-Cylinder Simple Superheater Locomotive. Engineer, vol. 135, no. 3504, Feb. 23, 1923, pp. 200–201, 3 figs. 2-6-0-type engines, built at Ashford works to design of R. E. L. Maunsell. See also Ry. Gaz., vol. 38, no. 8, Feb. 23, 1923, pp. 282–283, 3 figs.

MEASURING INSTRUMENTS

Small Motions. Precision Measuring Instrument for Small Motions of Solid Bodies, H. A. Thomas. Engineer, vol. 135, no. 3502, Feb. 9, 1923, pp. 138–140, 12 figs. Apparatus consists essentially of high-frequency electrical oscillator in which amplitude of oscillation is varied by motion of body under observation.

MOTOR TRUCKS

American Specifications. American Trucks Approach Standard Design in Major Features. Automotive Industries, vol. 48, no. 8, Feb. 22, 1923, pp. 402–419, 6 figs. Specifications indicate only slight changes over last two years. Specifications for gasoline trucks and motor buses.

Axle, Triple Reduction. Triple Reduction Features New 5-Ton L M Truck Axle. Automotive Industries, vol. 48, no. 9, Mar. 1, 1923, pp. 518–519, 2 figs. Said to be more compact than one of double-reduction type with same ratio; differential placed on highest-speed shaft where stresses are lowest; first and third reductions by spur gears, second by bevels.

OIL ENGINES

Heavy Oil Engines, Evolution of. The Evolution of the Heavy Oil Engine, R. E. Mathot. Engineer, vol. 135, no. 3502, Feb. 9, 1923, pp. 137–138. Comparison of four-stroke explosion motors with Diesel engines operating on same cycle. Points out advantages of moderate pressures applied to engines without use of compressed air for fuel pulverization.

OPEN-HEARTH FURNACES

Krupp Plant. Krupps Build New Open Hearths. Iron Trade Rev., vol. 72, no. 9, Mar. 1, 1923, pp. 661–663, 4 figs. Plant started during war and recently remodeled one of most complete in Europe; cold-metal process used now, but provisions are made for molten-metal process in future.

PUMPS, CENTRIFUGAL

Extraction. A New Condenser Extraction Pump. Engineer, vol. 135, no. 3502, Feb. 9, 1923, p. 152, 2 figs. New type of high-vacuum centrifugal extraction pump placed on market by Mirrlees Watson Co., which will form one of integral units of Mirrlees closed-feed system; chief feature is elimination of all air leakage. See also Iron & Coal Trades Rev., vol. 106, no. 2867, Feb. 9, 1923, p. 185, 4 figs.

RAILWAY ELECTRIFICATION

Chicago, Milwaukee & St. Paul Ry. Railway Electrification, Arthur L. Mudge. Eng. Jl. (Eng. Inst. Can.), vol. 6, no. 3, Mar. 1923, pp. 127–133, 5 figs. Notes on electrification of Chicago, Milwaukee & St. Paul Ry. Comparison with steam operation; transmission lines and substations; locomotive inspection and maintenance.

German Report. The Electrification of the German Railways. Engineer, vol. 135, no. 3503, Feb. 16, 1923, p. 169. Excerpt of report of sub-committee appointed by Assn. German Ry. Administrations, containing information on question of system and proposed standards. Notes on Berlin, Austrian Federal and Dutch railways.

RAILWAY SIGNALING

Interlocking. Signaling Increases Capacity of Three Tracks. Ry. Age, vol. 74, no. 9, Mar. 3, 1923, pp. 498–501, 6 figs. Automatic signaling and interlocking plants on 21-mi. stretch of third track nearing completion on Ill. Central; center track signaled both ways, with interlocking, allows fast trains to pass slow ones.

RAILWAY TIES

Preservative Treatment. Creosote Shortage Threatens Wood Preservation, C. M. Taylor. Ry. Age, vol. 74, no. 9, Mar. 3, 1923, pp. 505–507, 3 figs. Mixing crude oil with creosote, it is claimed, will increase supply and provide necessary protection.

RAILWAYS

Engineering Problems. The Railroad Engineer and the Needs of Tomorrow, W. S. Kinnear. Eng. News-Rec., vol. 90, no. 10, Mar. 8, 1923, pp. 428–431. Survey of problems of railway development and plea for policy that will help engineer to solve them.

SCREW MACHINES

High-Speed Automatic. Screw Machine Products Made in Less than Three Seconds, Luther D. Burlingame. Machy. (N. Y.), vol. 29, no. 7, Mar. 1923, pp. 507–511, 9 figs. Examples showing how rapid production is obtained by careful tooling and machine timing. High-speed automatic screw machines of Brown & Sharpe Mfg. Co.

STANDARDIZATION

Germany. Standards in Germany Industry, George E. Hagemann. Management Eng., vol. 4, no. 3, Mar. 1923, pp. 183–188, 4 figs. Total number adopted to date is 870 and about 15 are added monthly. Organization of German Industrial Standards Committee; working up a standard; similarity to American method. Table of standards adopted up to January, 1923.

STEAM-ELECTRIC PLANTS

Ford Plant, River Rouge. Ford Principles and Practice at River Rouge, John H. Van Deventer. Indus. Management (N. Y.), vol. 65, no. 3, Mar. 1923, pp. 149–160, 28 figs. Development and operation of power plant.

STEEL

Tensile Tests. Tensile Tests of Materials at High Temperatures, F. C. Lea. Engineer, vol. 135, no. 3503, Feb. 16, 1923, pp. 182–183, 3 figs. Apparatus used; effect of temperature on fatigue range of stress static tests of nickel-chrome and high-carbon steels at high temperatures. (Abstract.) Paper read before Junior Instn. Engrs.

STRESSES

Repeated, Failure under. The Effect of Repetition Stresses on Materials, F. C. Lea. Engineering, vol. 115, nos. 2981 and 2982, Feb. 16 and 23, 1923, pp. 217–219 and 252–254, 15 figs. Experimental study; results of endurance tests; optical and thermal methods of determining fatigue ranges. Paper read before Instn. Civ. Engrs.

THERMOMETERS

Mercury. The Exposed Stem Correction for Mercury Thermometers, Wm. L. DeBaufre. Power, vol. 57, no. 9, Feb. 27, 1923, pp. 320–321, 4 figs. Measuring temperature of exposed stem; ordinary and direct formula for correction; method of constructing large-scale chart for direct formula.